# AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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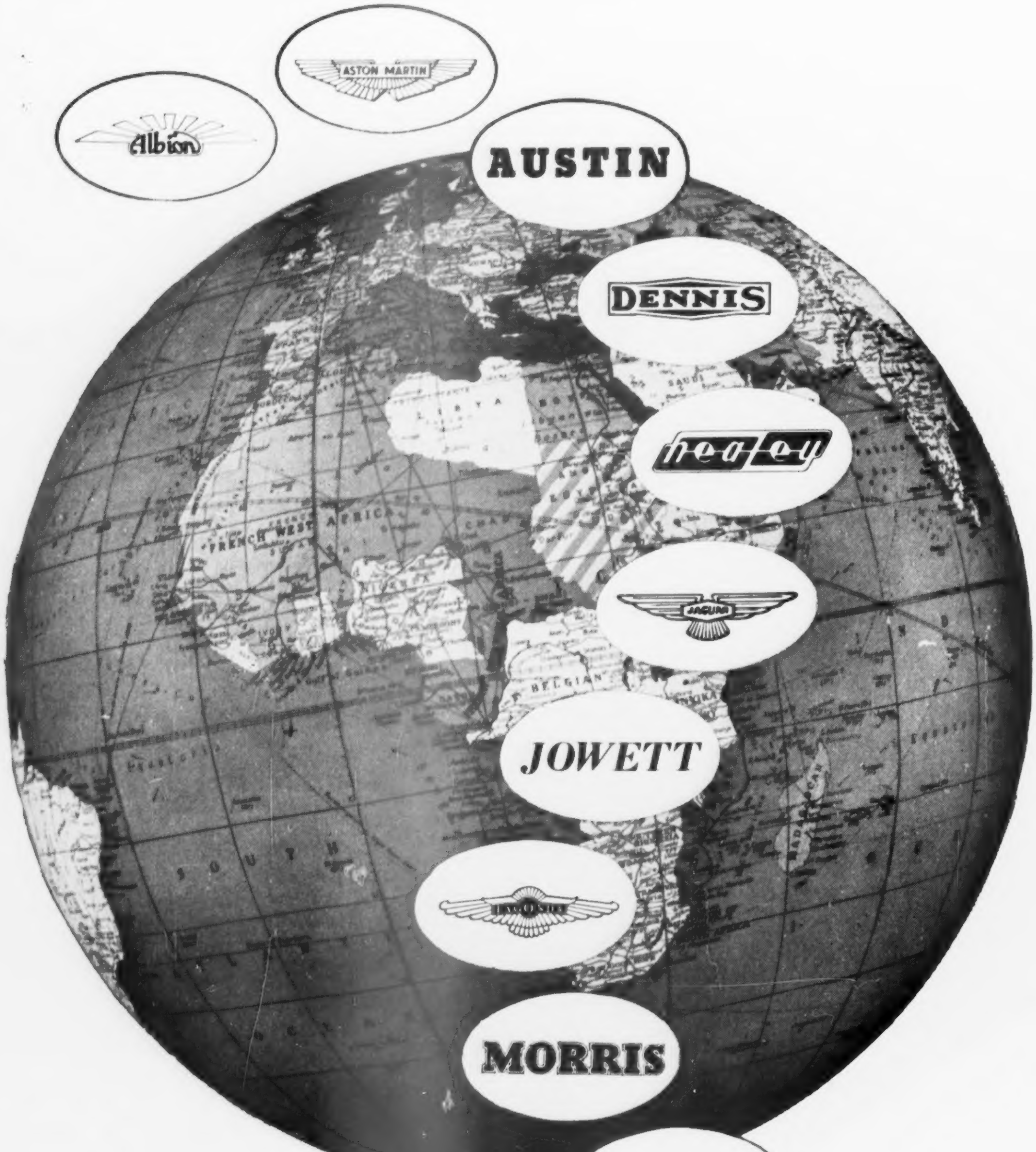
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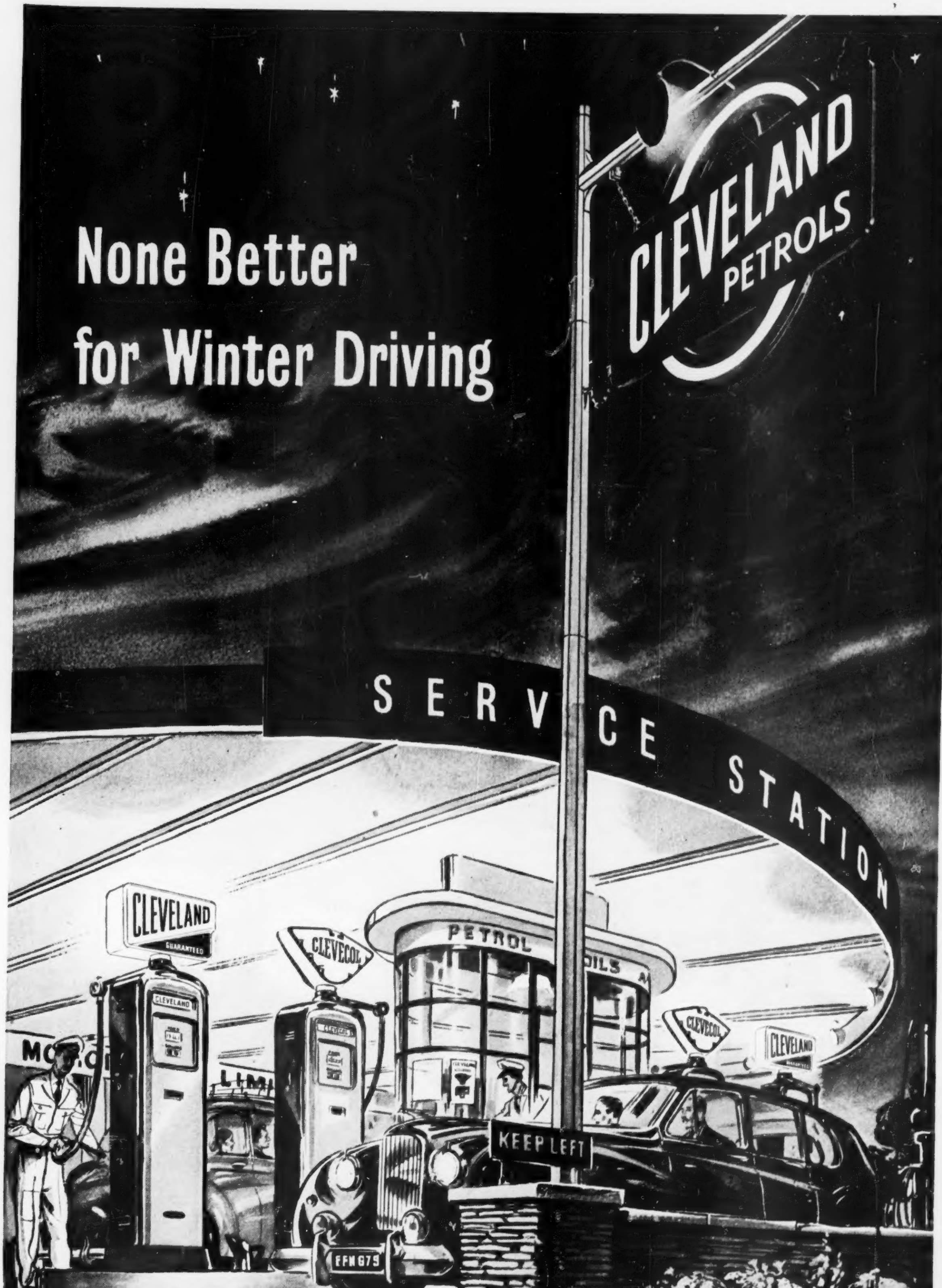
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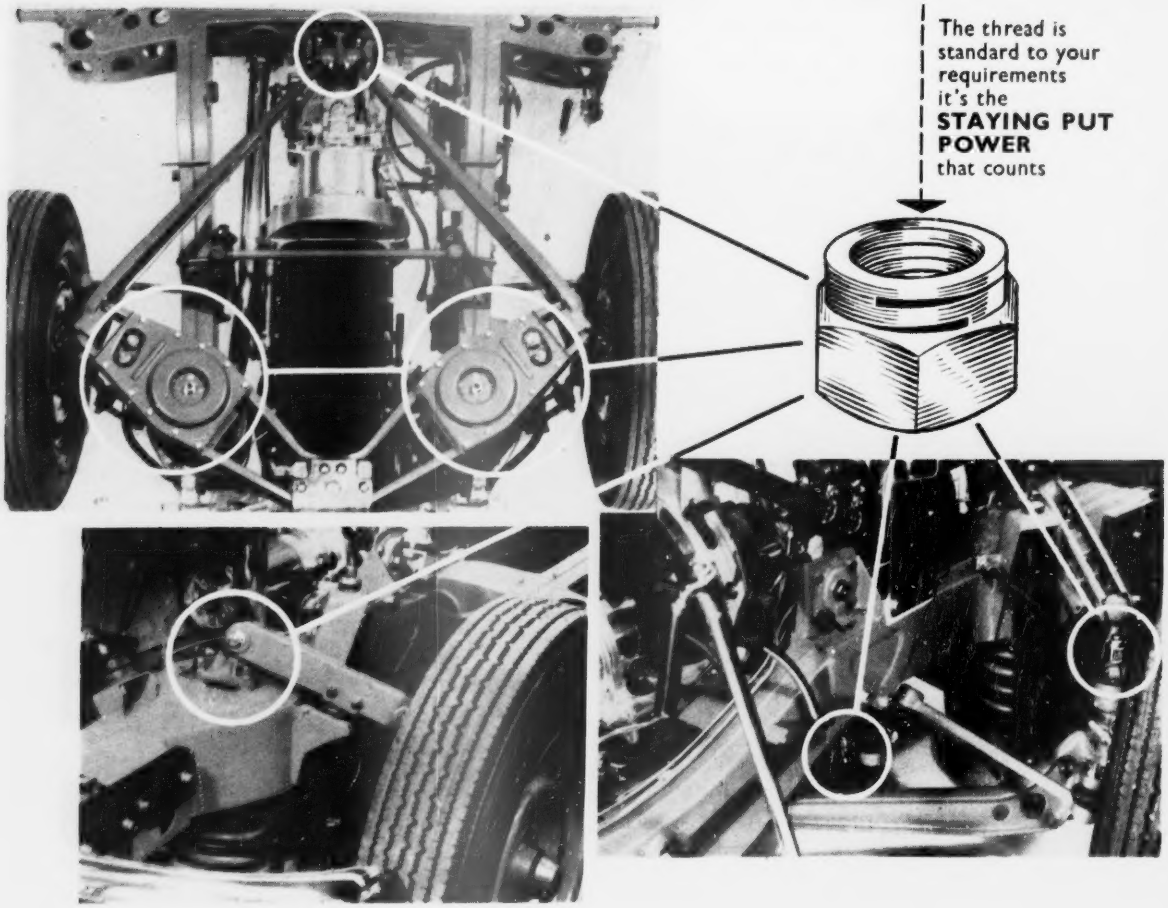


These photographs reproduced by courtesy of the Rover Co. Ltd., show Philidas nuts in position on a sectioned chassis section of the Rover car. Like many other leading motor manufacturers, the Rover Co. use Philidas nuts because :

They are immovable except with a spanner.

Their efficiency is unaffected by heat, vibration, oil contamination or by repeated applications and removals.

They are manufactured entirely from metal in one piece, and in all the usual thread types.

# PHILIDAS self-locking nuts

**STAYING PUT on the**

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*One of Britain's Fine Cars*

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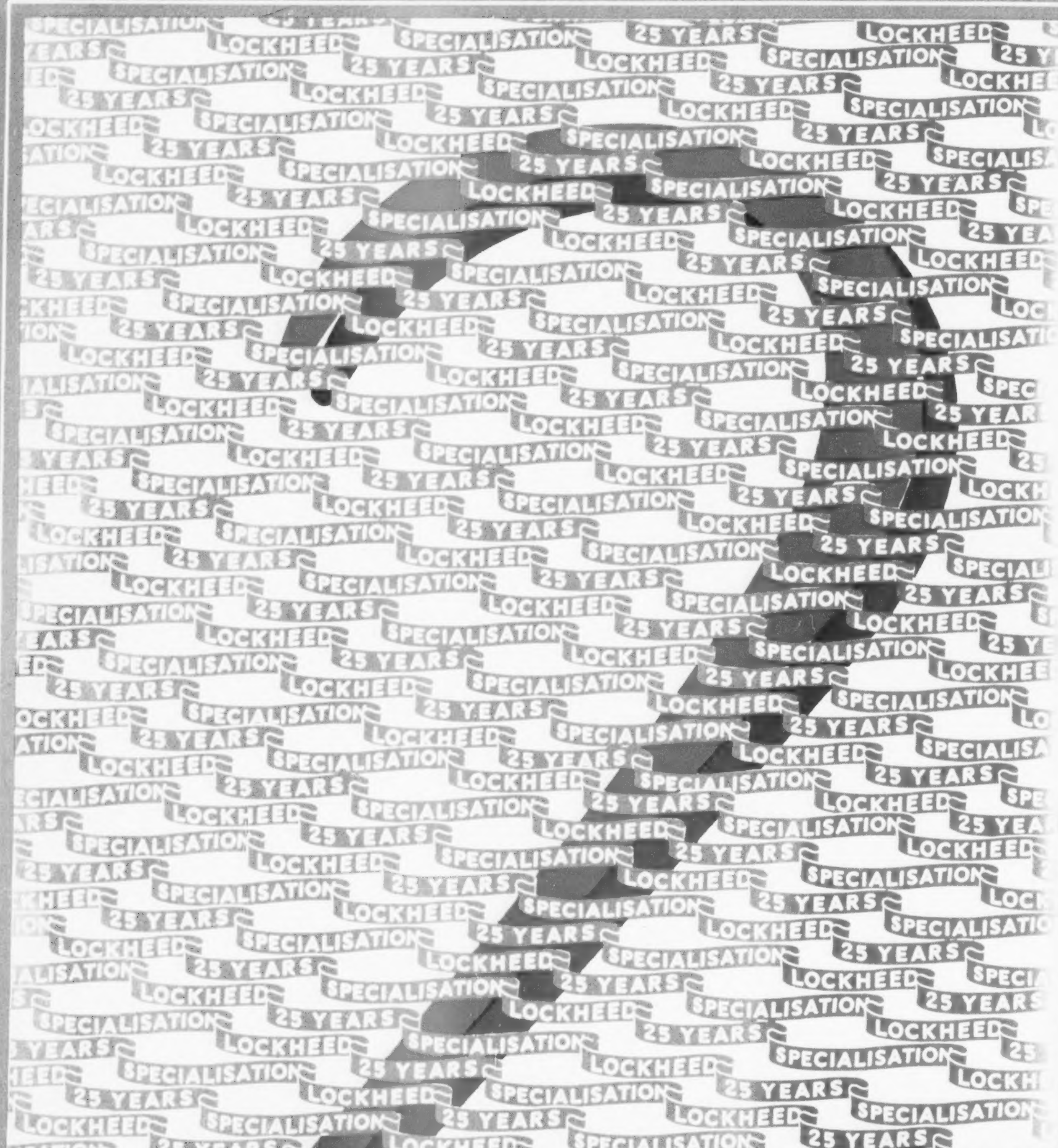
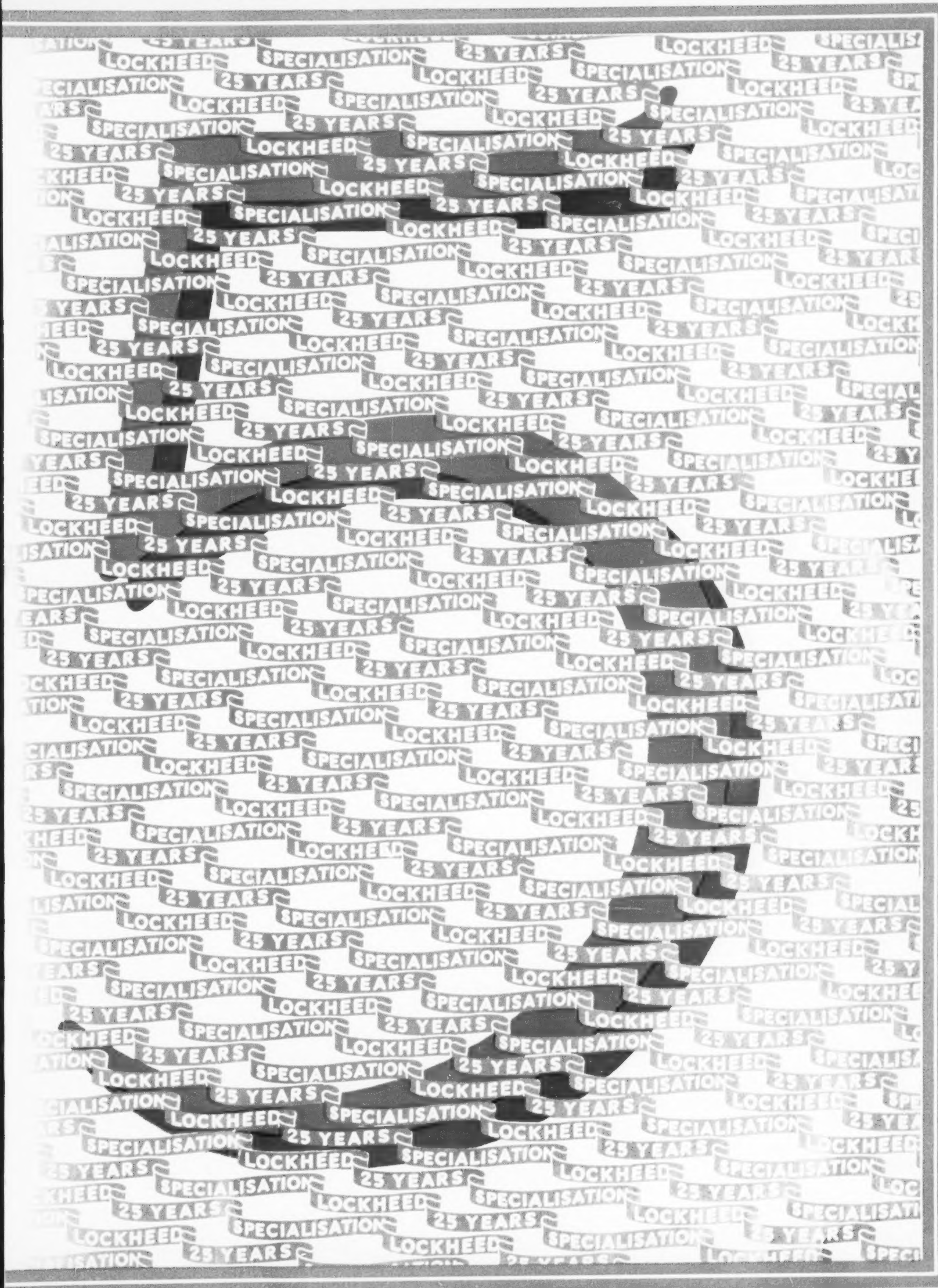



LOCKHEED HYDRAULIC BRAKES 25 YEARS OF SPECIALIZED PRODUCTION


LOCKHEED
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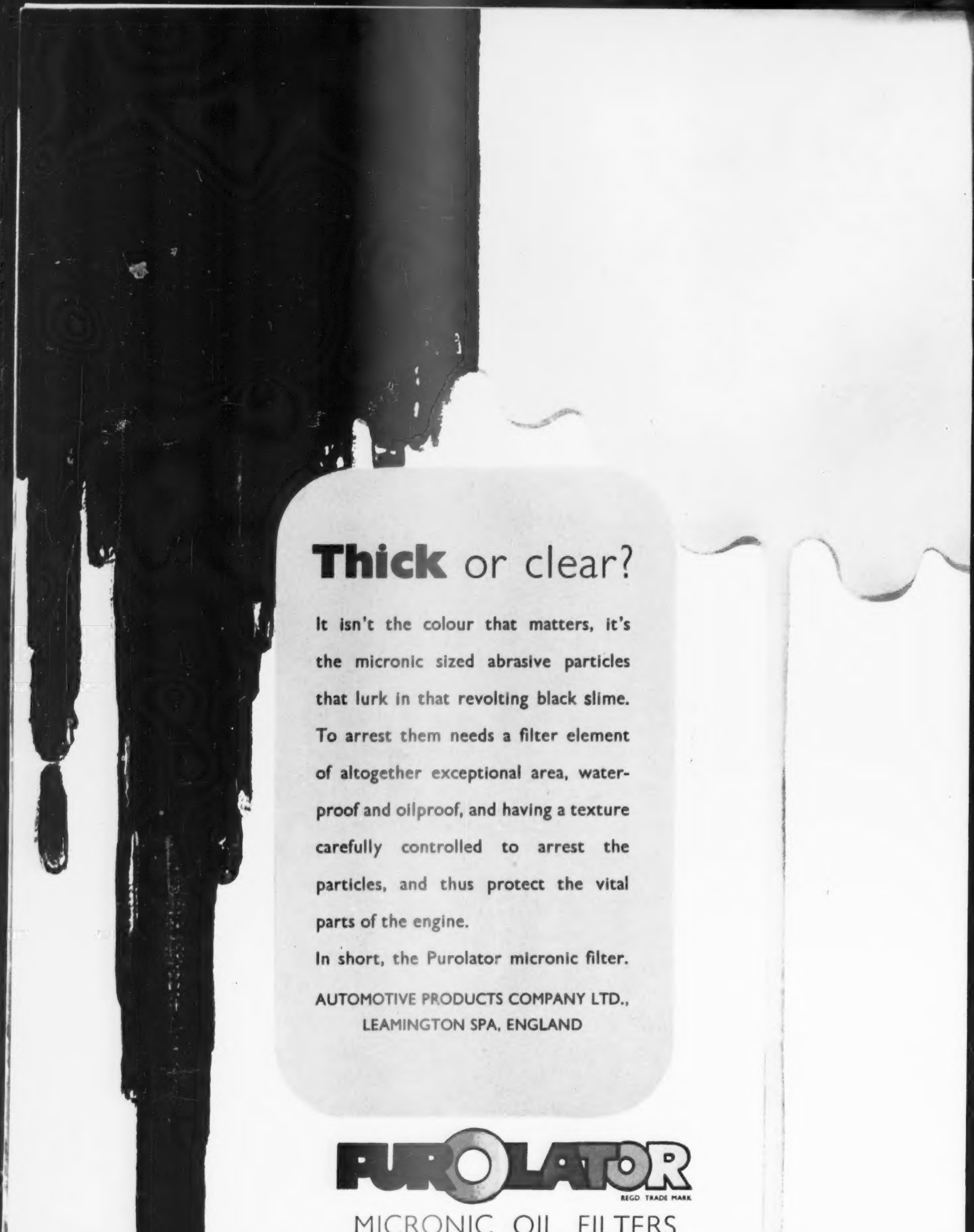
25 YEARS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In short, the Purolator micronic filter.
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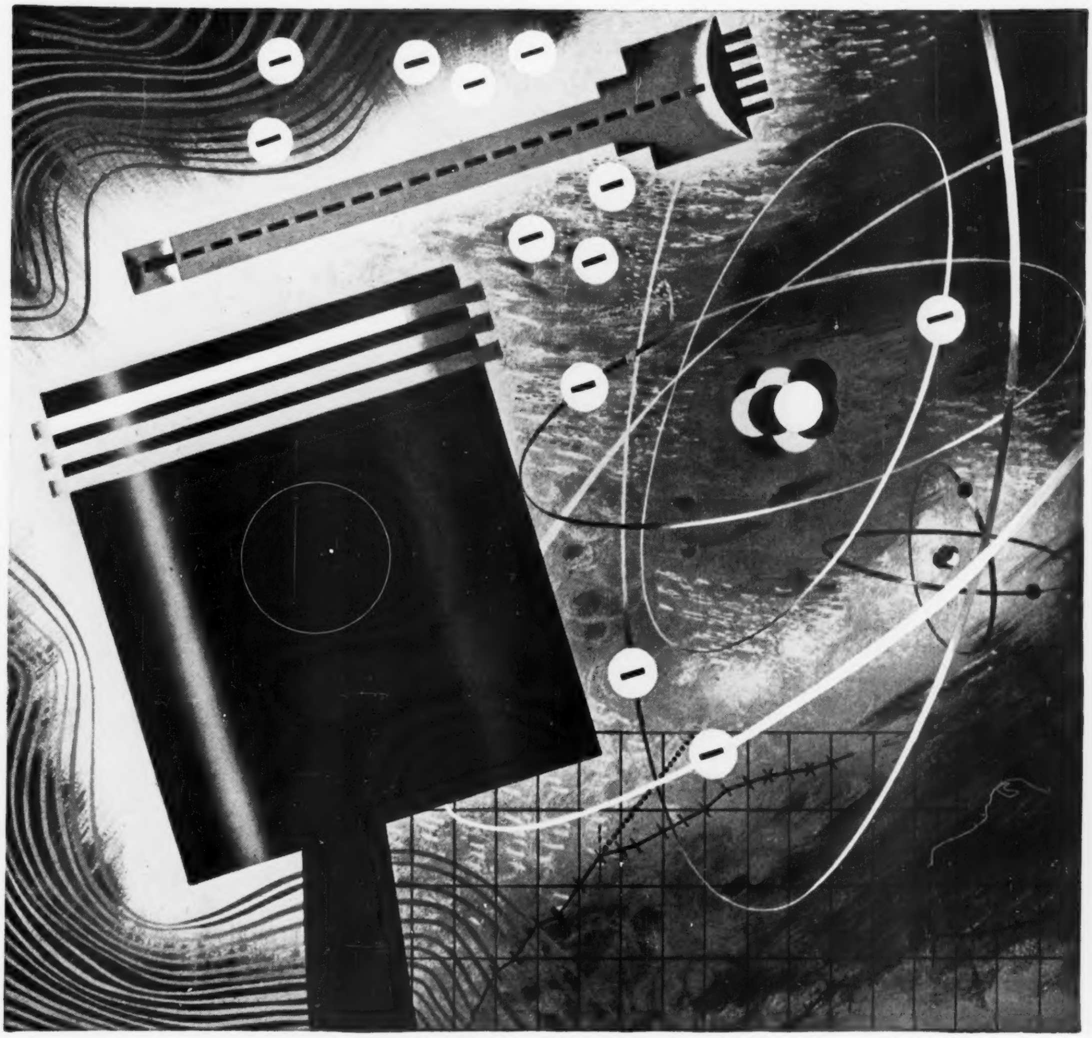



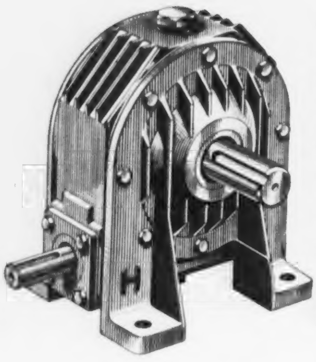


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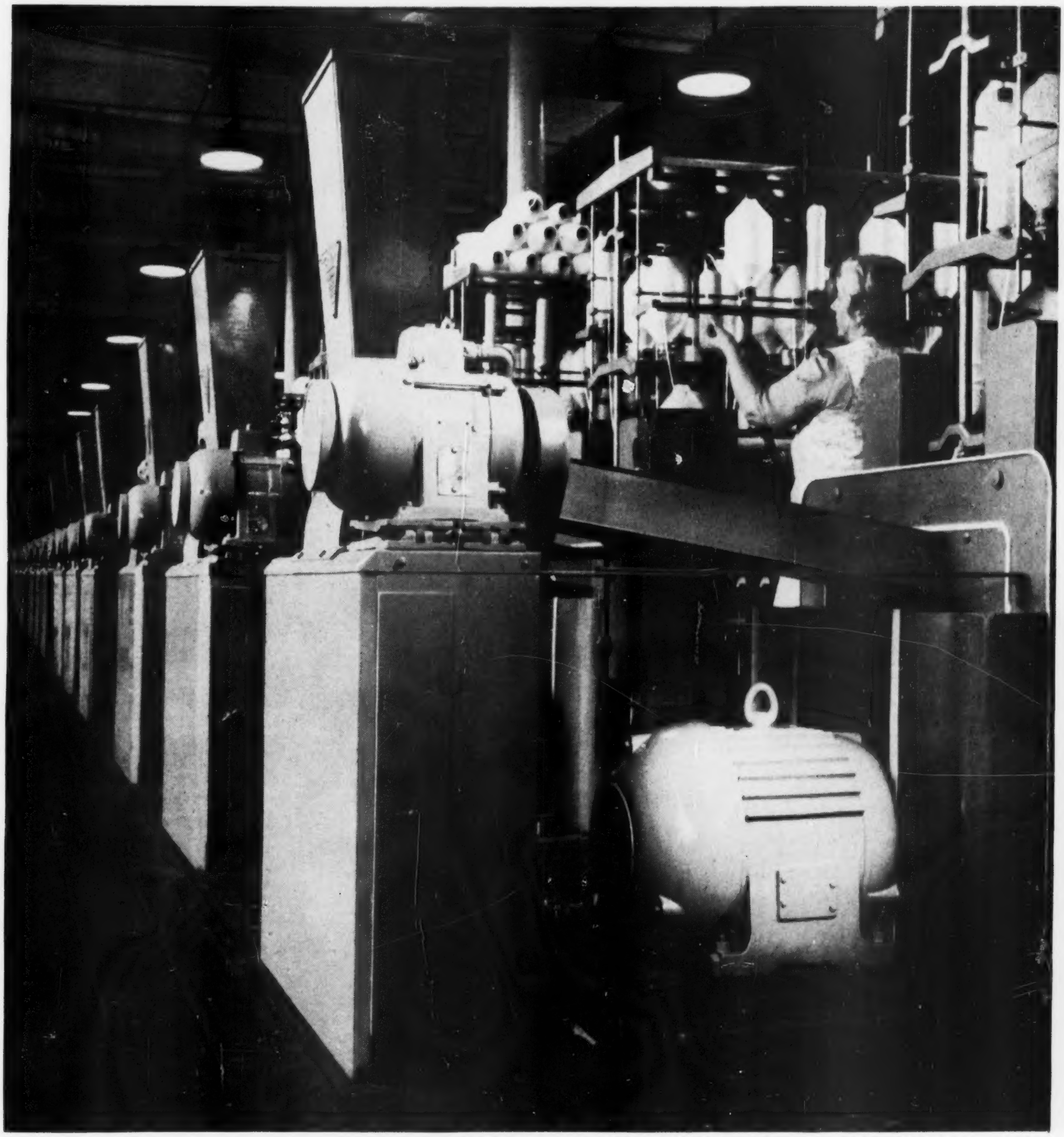

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**This rugged control device detects and is actuated by minute increases in motor current consumption due to dull or chipped cutting edges or clogging of flutes with chips.**

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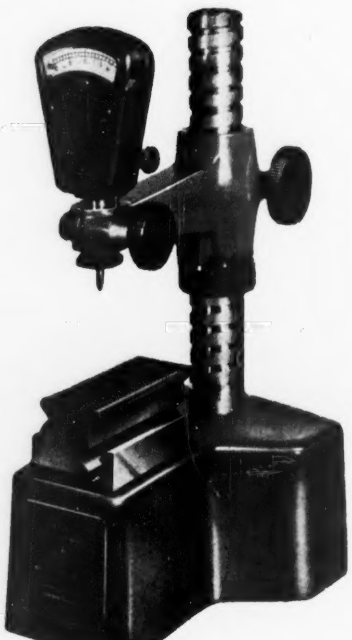

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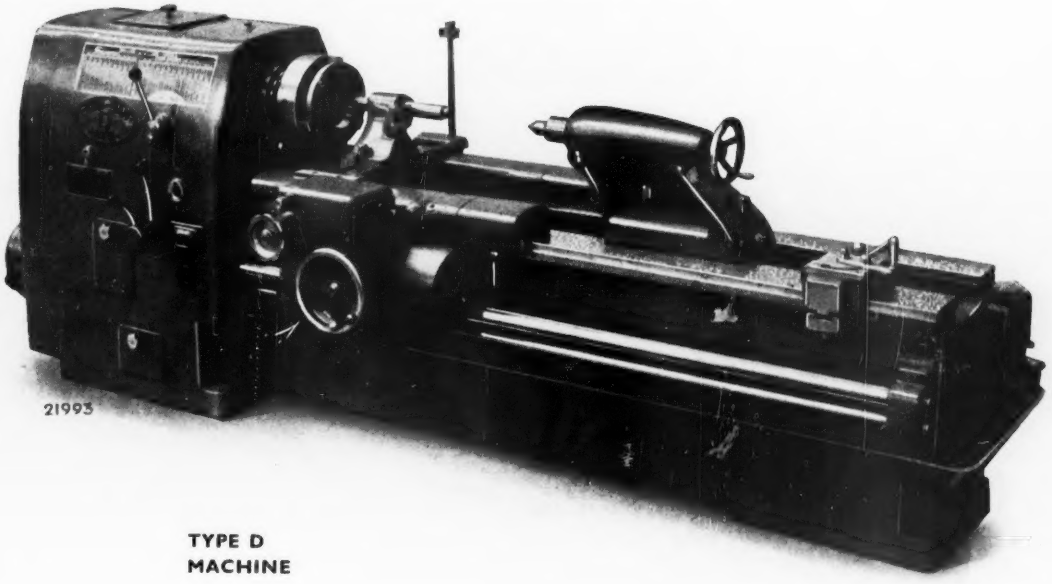


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These machines, which are semi-automatic, will cut external, internal, right- or left-hand, parallel, taper, single- or multi-start threads of various forms and pitches in a wide range of materials, quickly, cheaply and accurately.

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| Normal threading length | 2⅝" | 5½" |
| Length between centres | 3' | 5' |
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| Motor | 5 h.p. | 14 h.p. |

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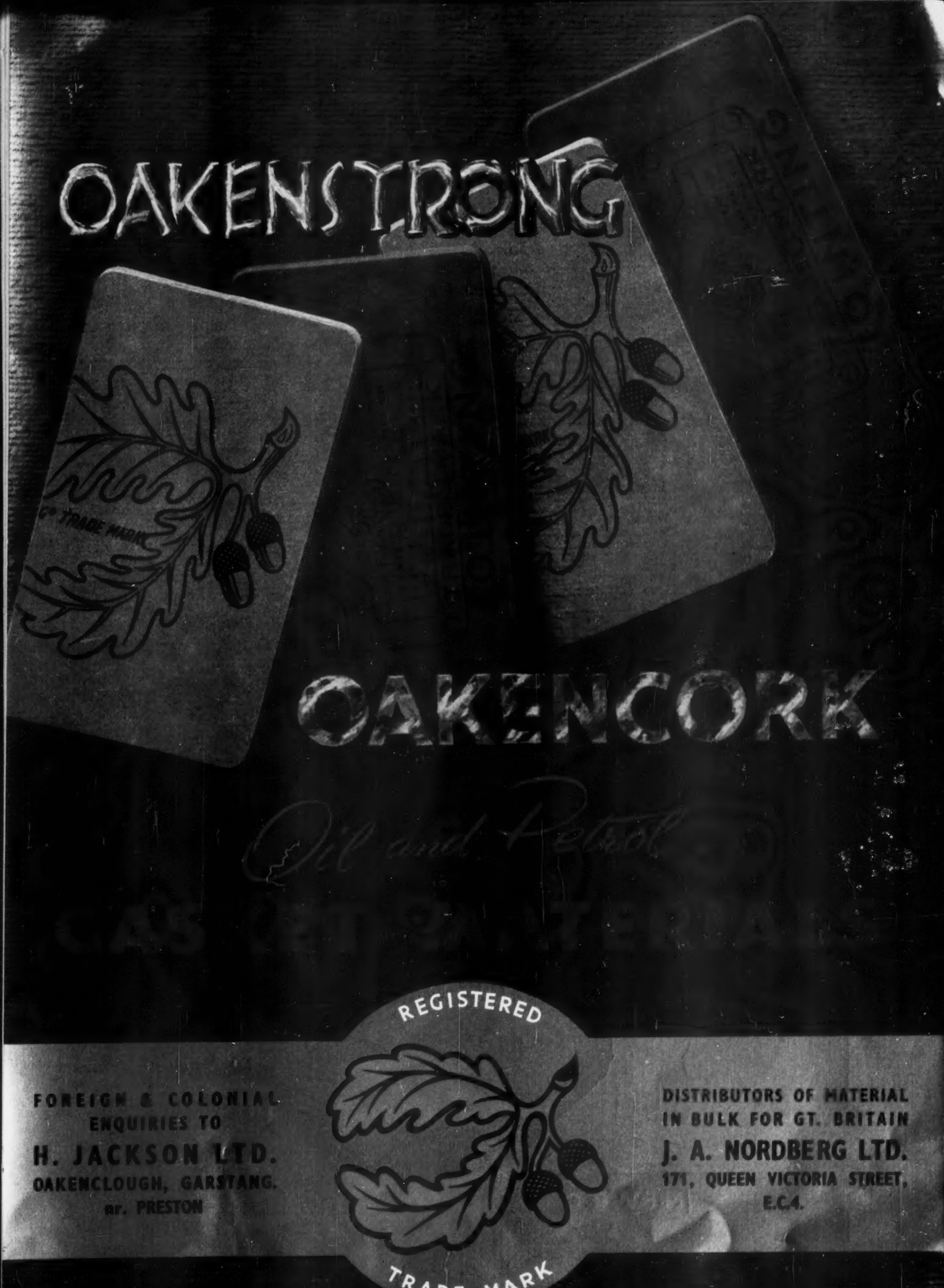

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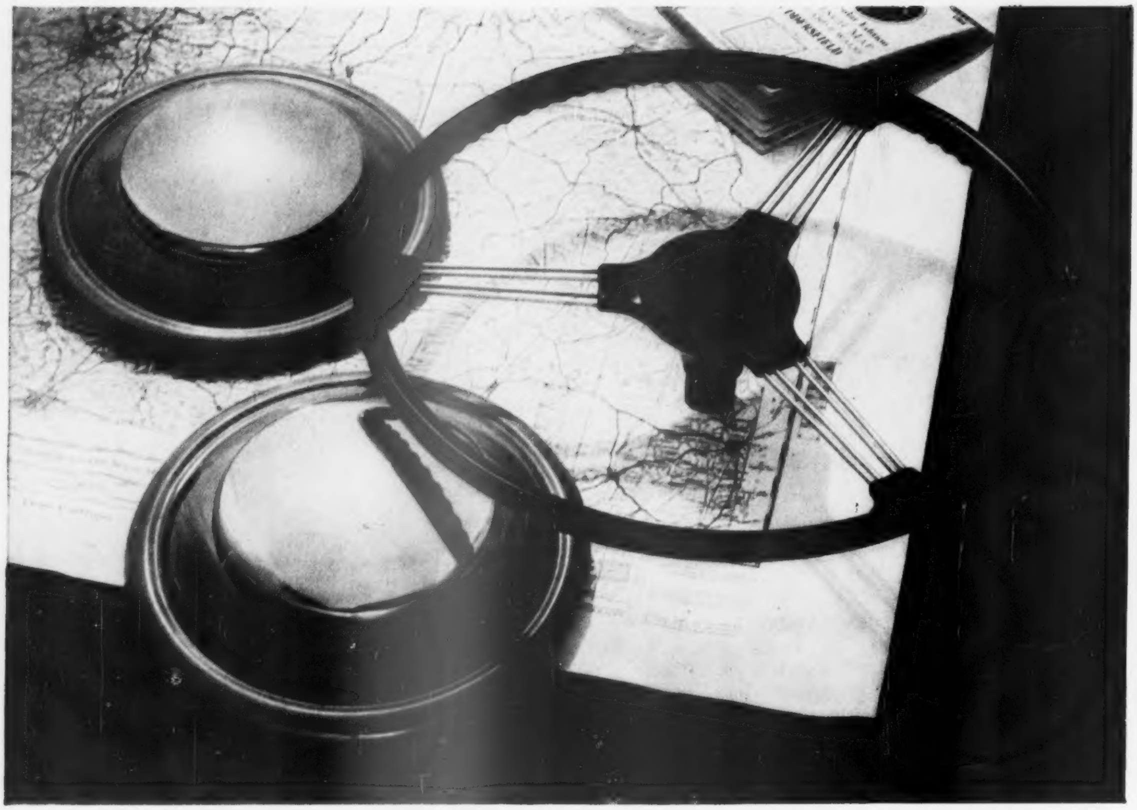
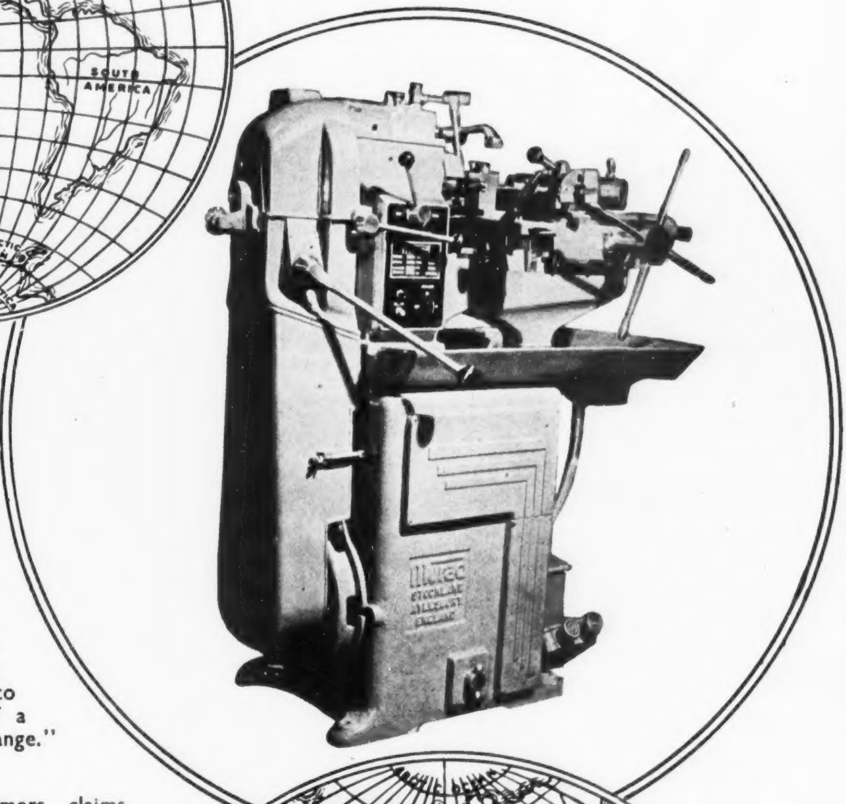



# *acclaimed in both hemispheres*



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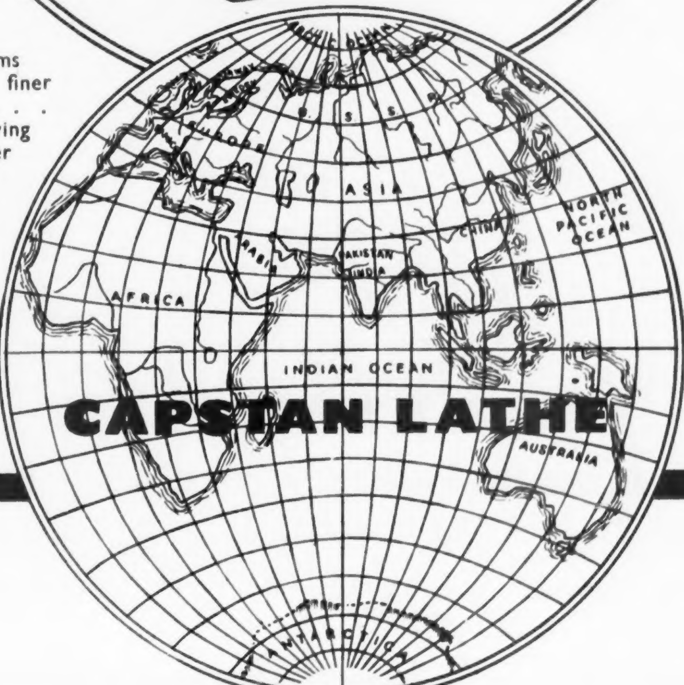


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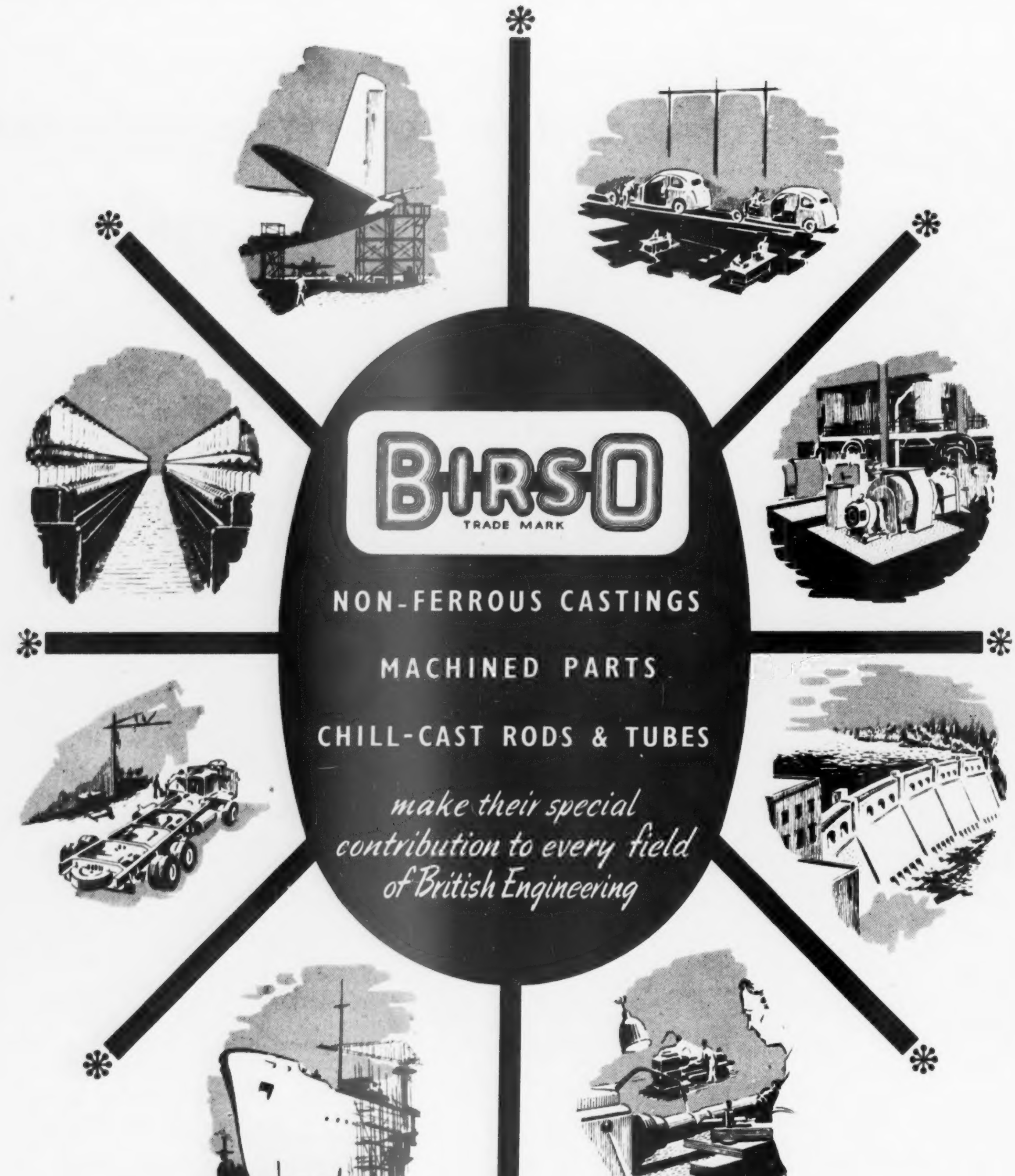
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Bob Gerard and his veteran E.R.A., with a Laystall crankshaft.


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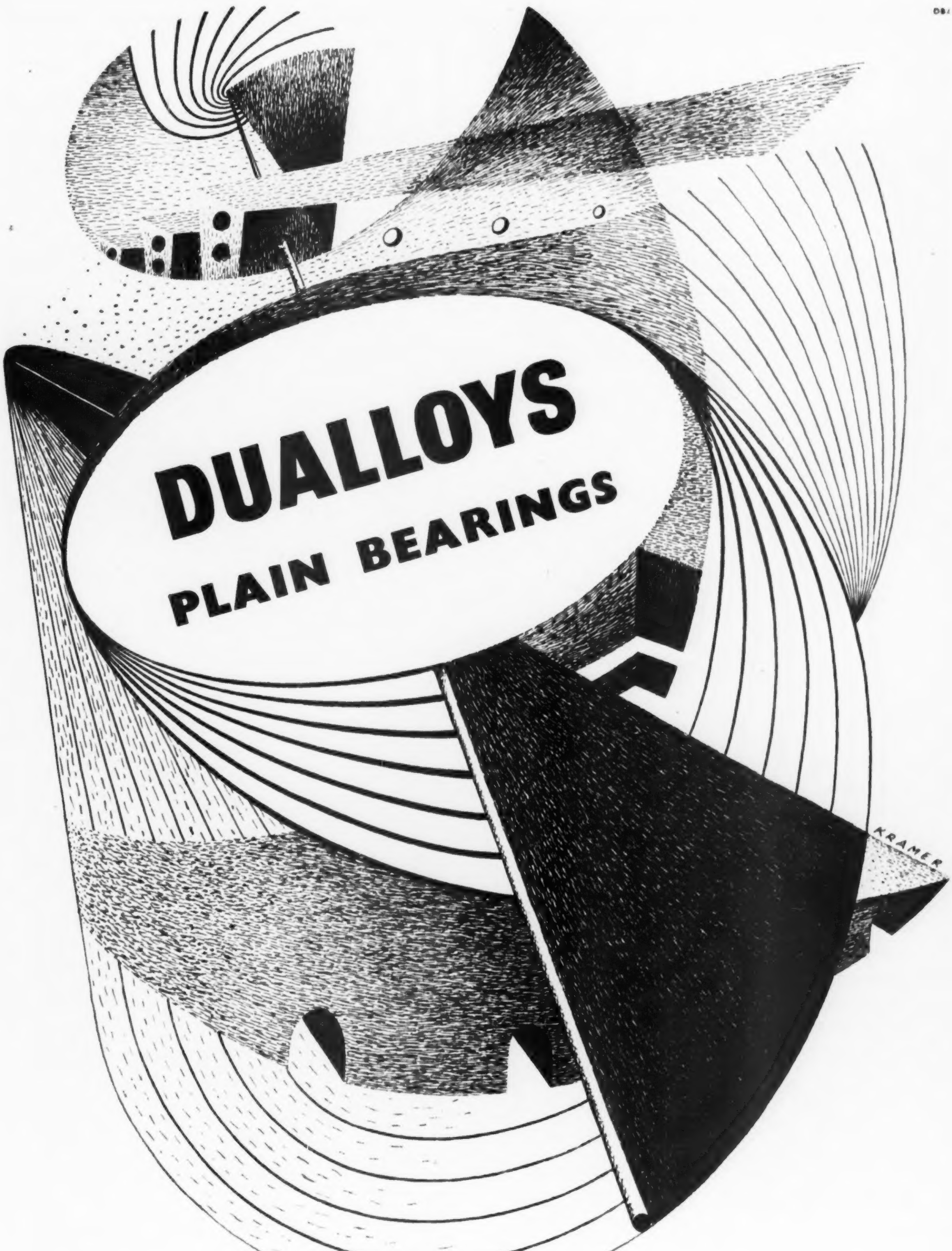
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This new AC partial-flow oil filter with replaceable element meets the increasing demand for a filter that can be removed for inspection and replaced every 8/10,000 miles. It provides a permanent installation for coupling into a by-pass oil supply system and the filtering element can be replaced without disturbing pipe connections. Sump oil is filtered on an average of ten times an hour. This means less motor wear and longer motor life — the objective of every motor manufacturer.

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**DIRECT - ACTING TYPE DAMPER**
**DAS Type.**
This type provides a light, compact, efficient and inexpensive means of damping out road shocks. Fluid displacement is large and the damper is designed for long life. It is available with a working stroke of from 3 to 10 inches. The diameter of the piston is 1 inch. Either stem or eye mounting can be supplied at either end as desired.

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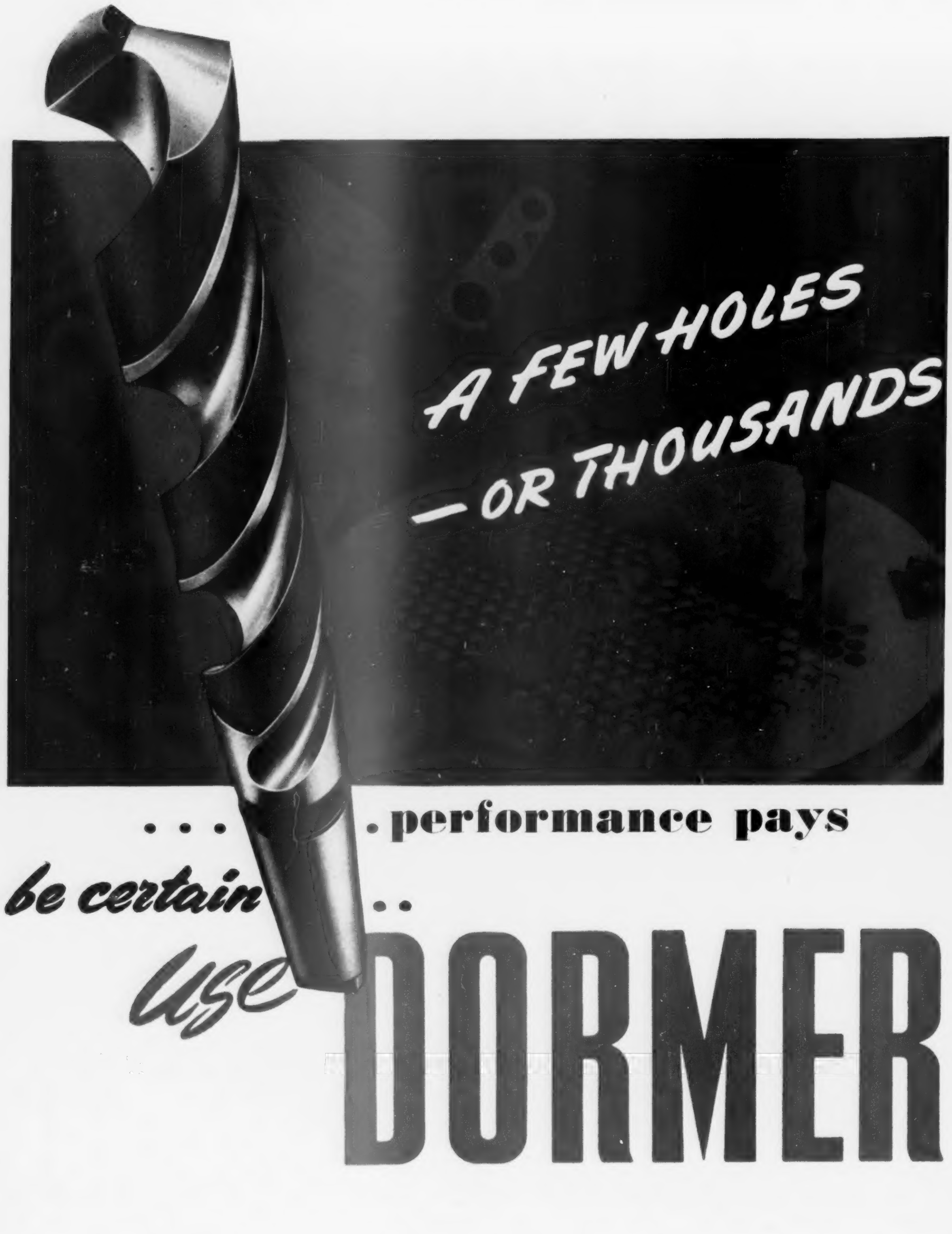
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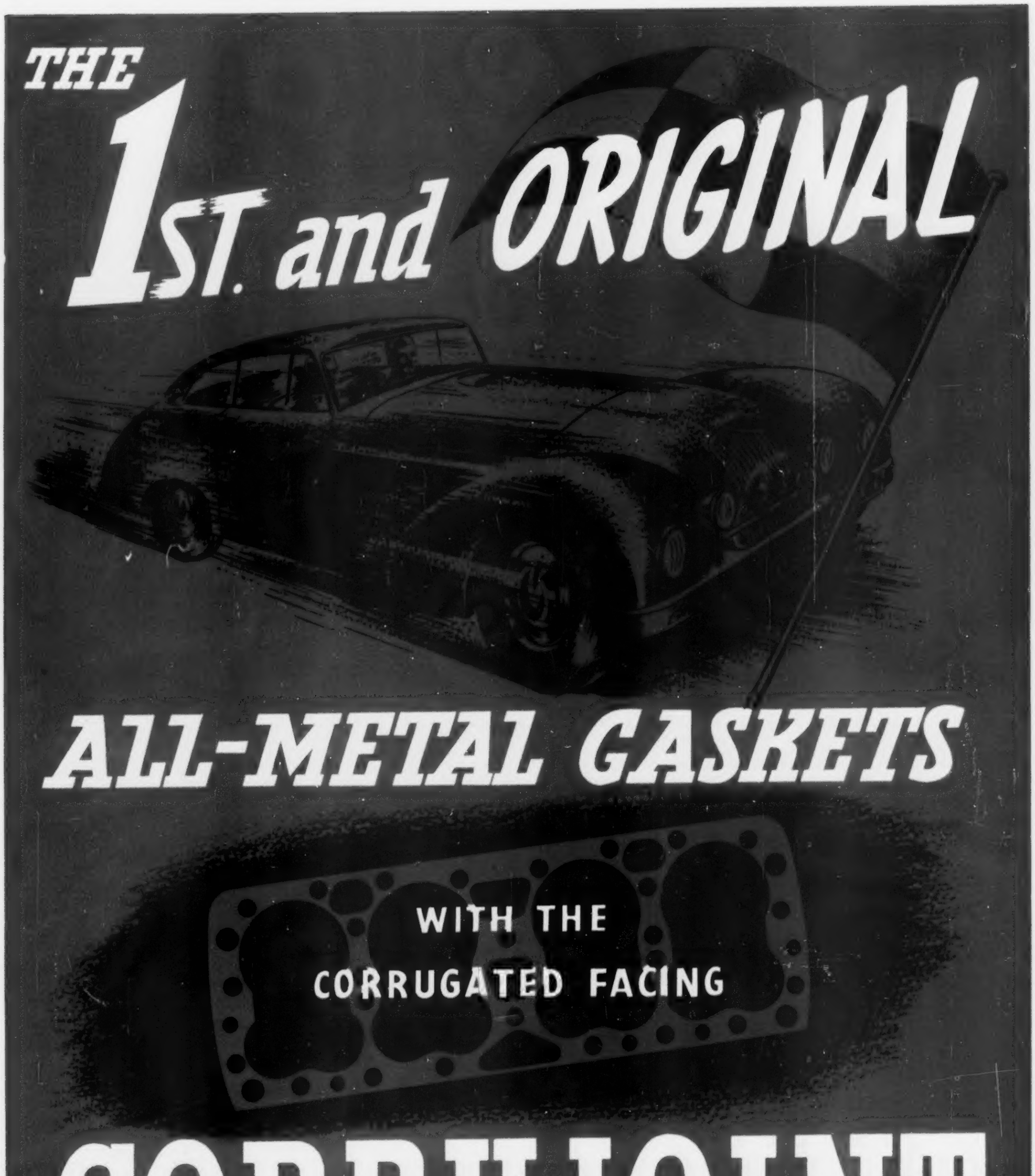
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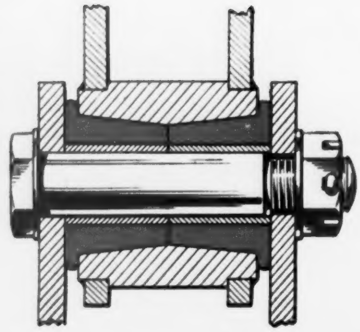


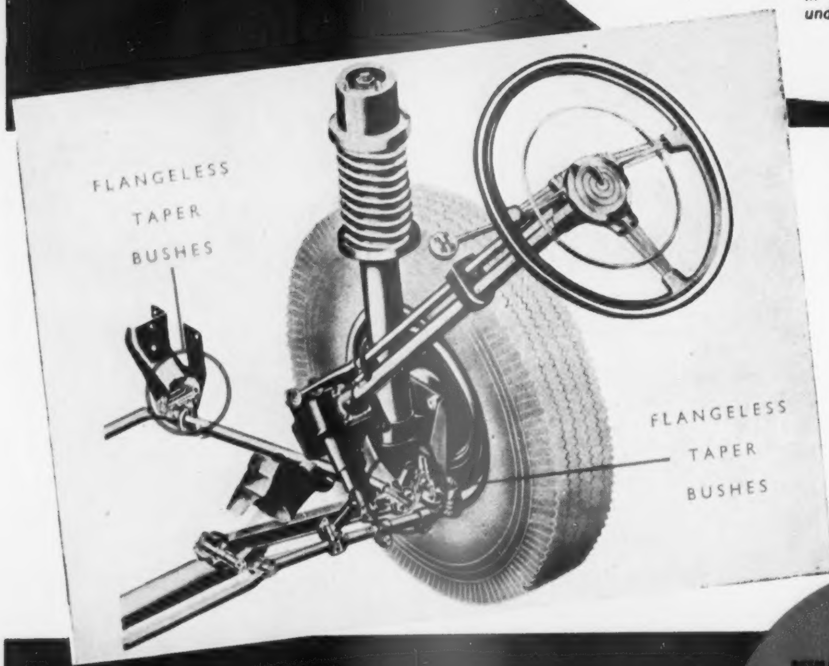
## FLANGELESS TAPER BUSHES

for I.F.S systems

### Scientific Development Cuts Costs

Constructed on the Silentbloc principle with a self-forming flange for extra-snug fitting and less wear these bushes ensure maximum efficiency at lowest cost.

*Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.*




★ After exhaustive testing Ford Motor Company Ltd. now fit Silentbloc Flangeless Taper Bushes in the I.F.S. of the "Consul" and "Zephyr Six".

*Left: Front Suspension of the "Consul" and "Zephyr Six".*

Flangeless Taper Bushes and Frustacon mountings provide the first complete and scientifically designed rubber insulation between wheel and body.

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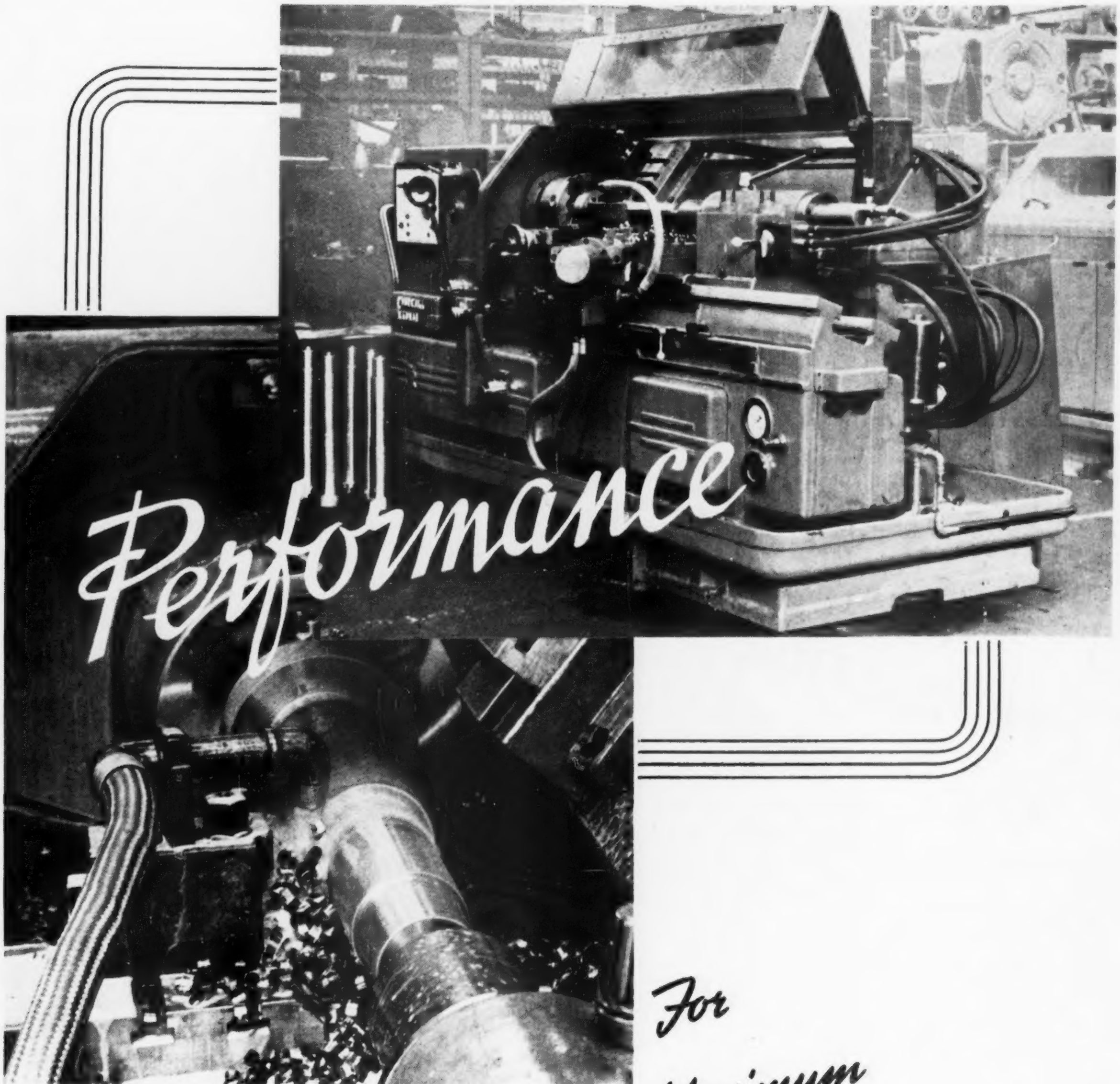
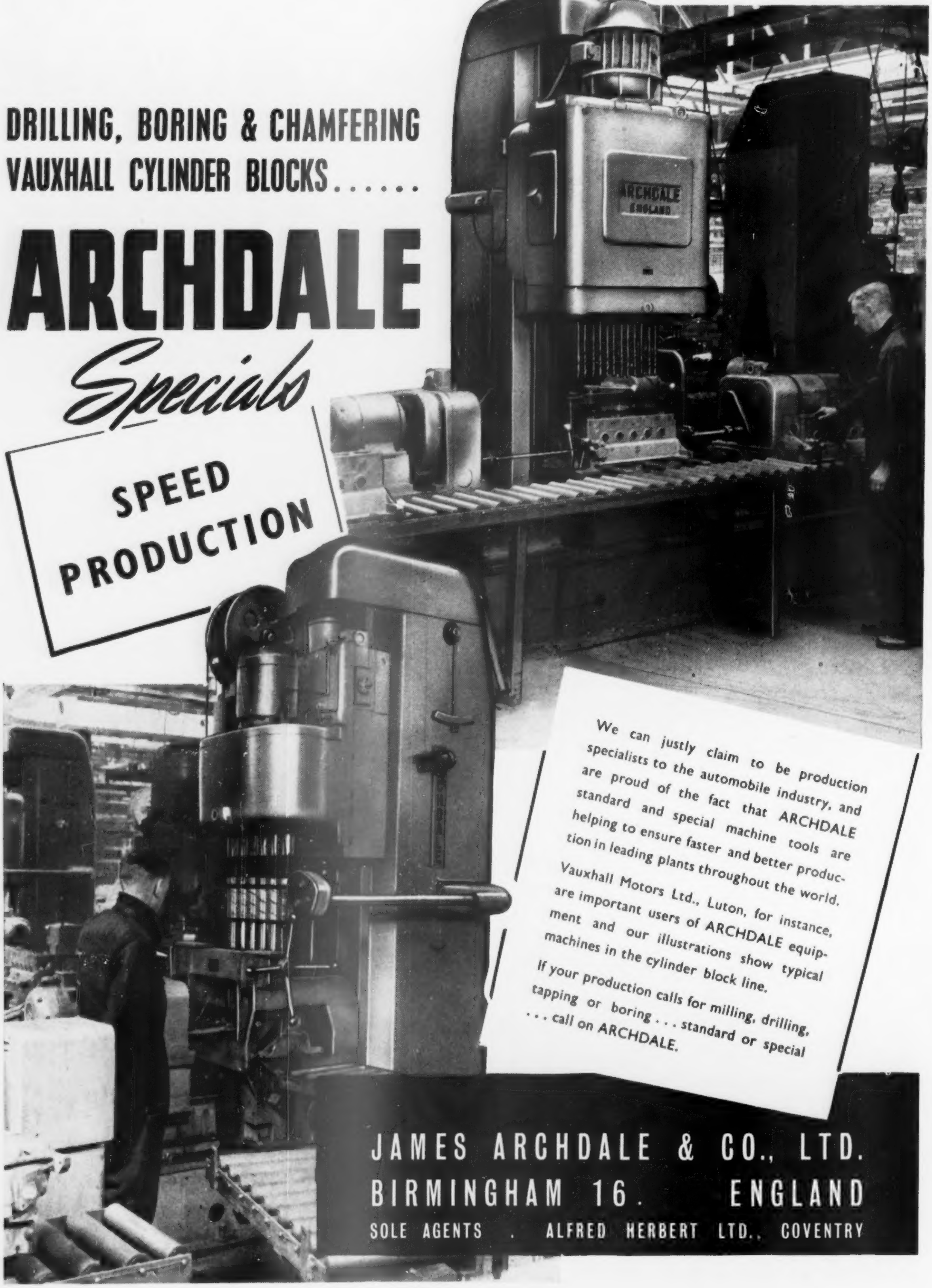

Performance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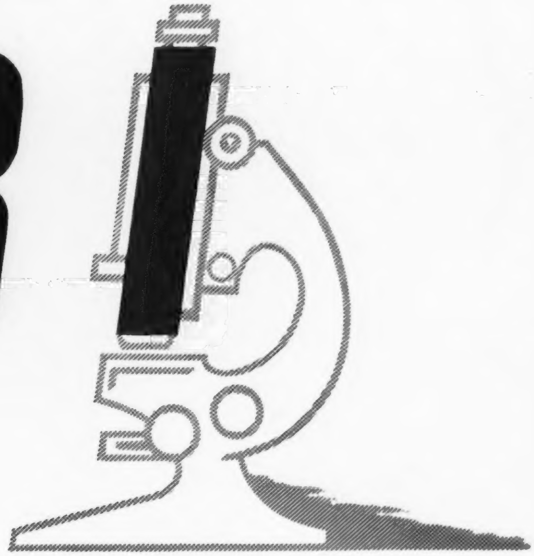

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From every batch of Ferodo friction materials, samples are taken and ruthlessly tested for efficiency and durability. On the M.I.R.A. Track (of which Ferodo Limited are one of the largest users) the final and practical proving of the finished product takes place. All this, and the ceaseless experimental work in the Ferodo Test House, ensures that Ferodo friction materials maintain their high standard of reliability, efficiency and durability that makes them first choice for safety all over the world.

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WE PERFECTED *bushes for I.F.S. to a higher pitch of perfection than all the other manufacturers in the world put together.*

WE DEVELOPED the Metalastik torsional *vibration damper—'tunable,' sensitive, free from metal-to-metal contacts, and with an all-round superiority to other types of crankshaft dampers.*

**METALASTIK**

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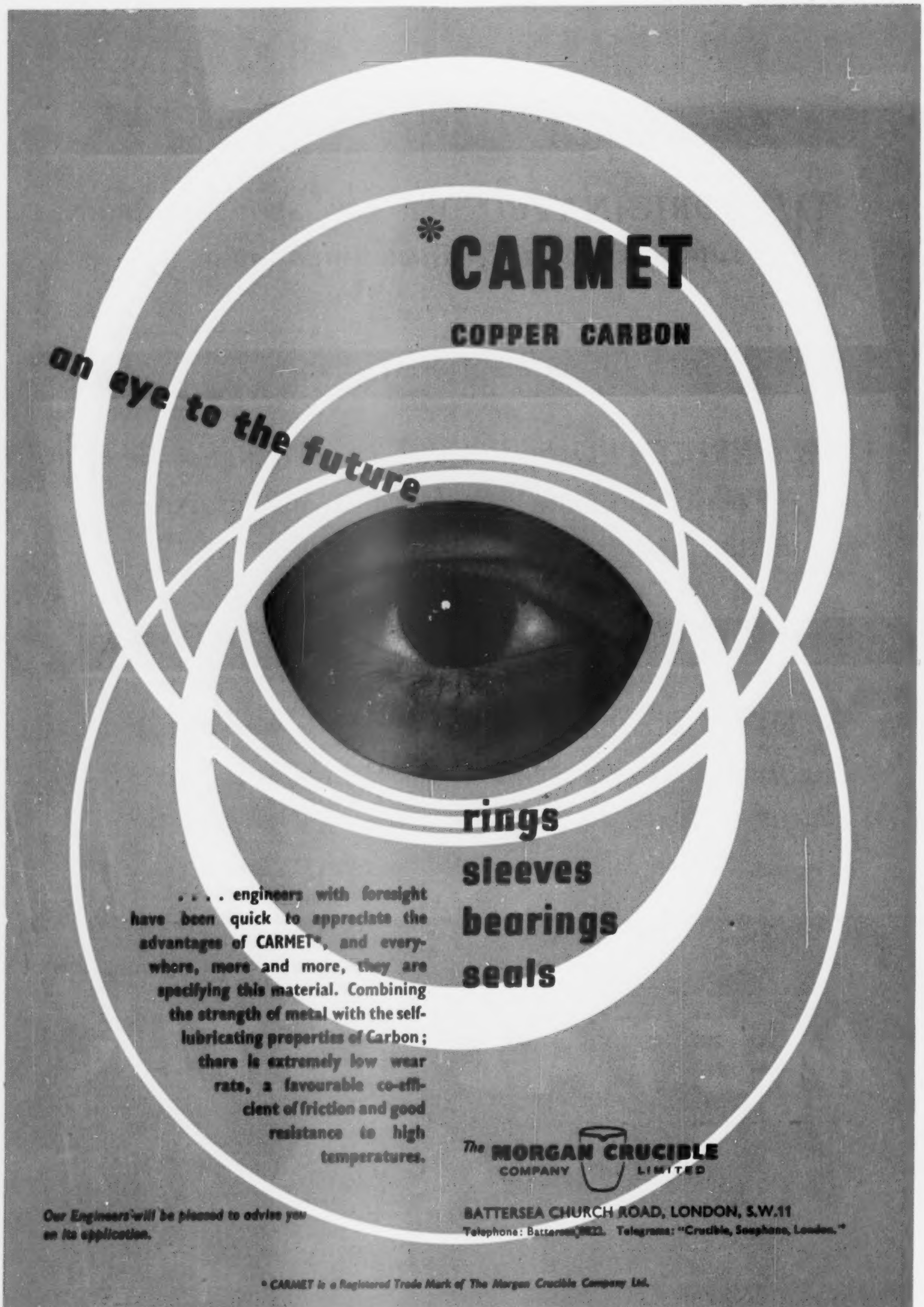
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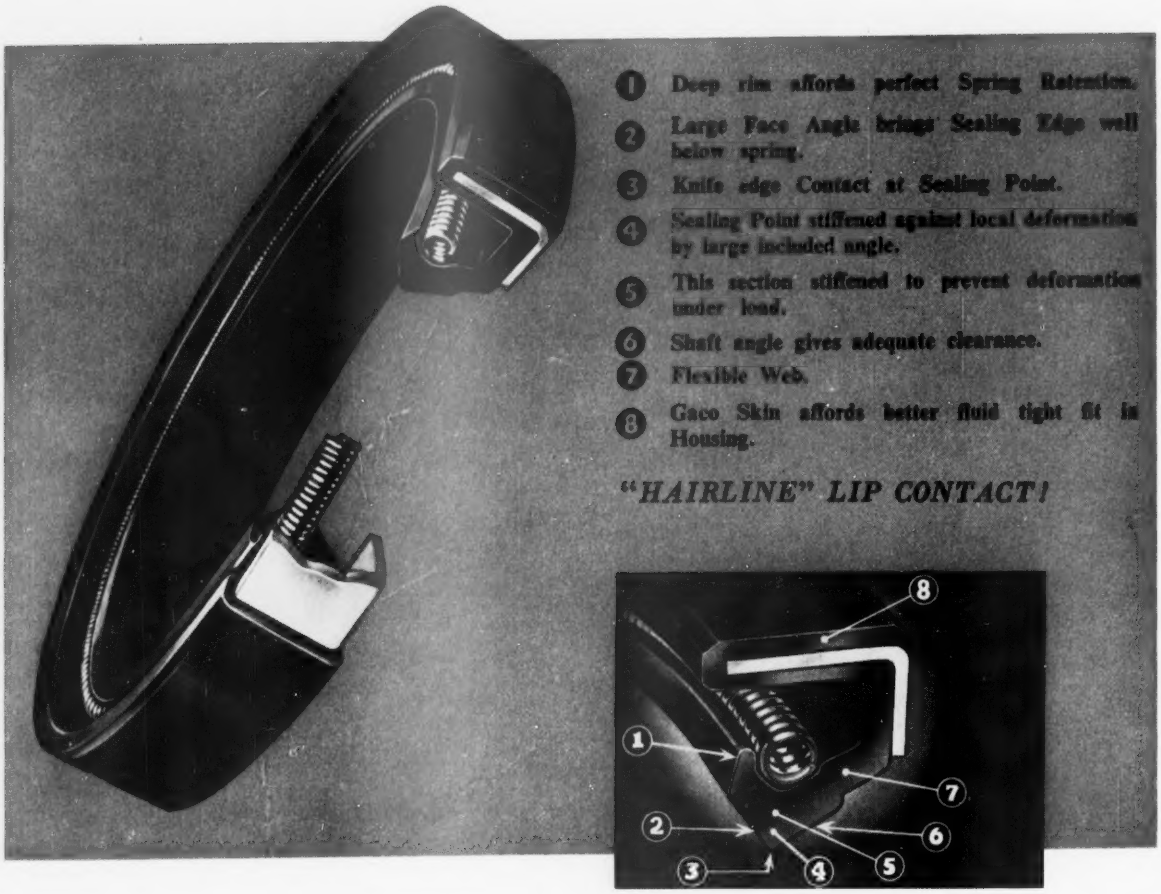
# THE GACO M.I. OIL SEAL

British Patent No. 479743

*The Scientifically Designed Oil Seal for Rotary Shafts. A Gaco Product.*

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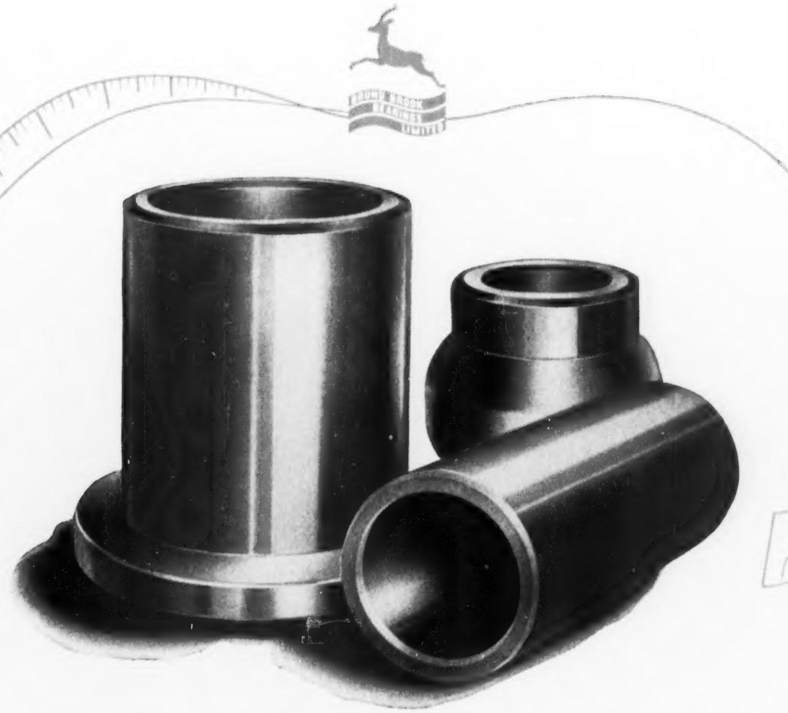

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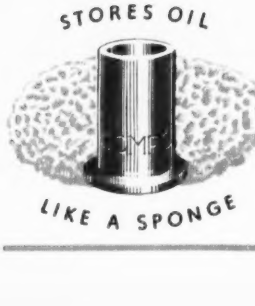


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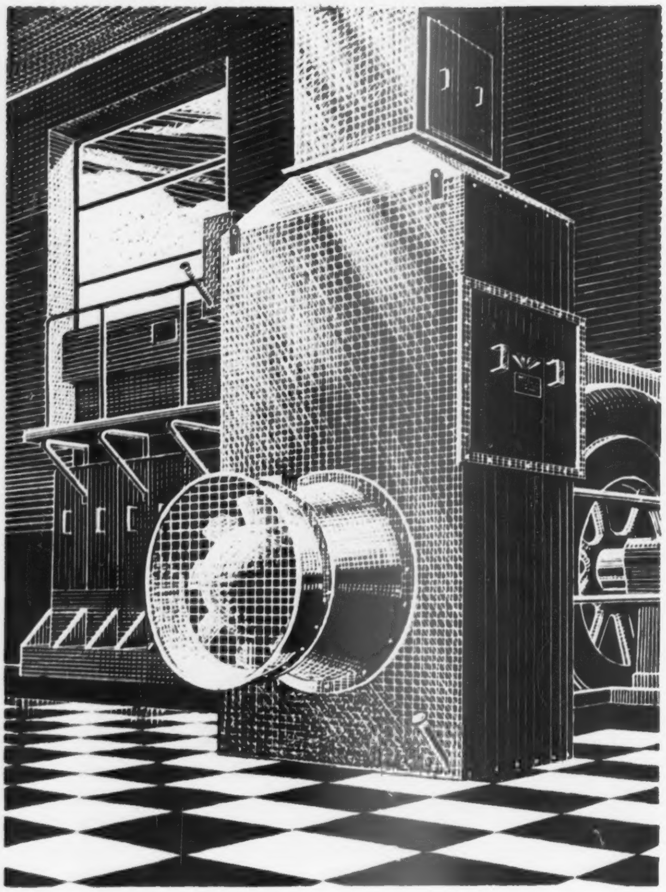

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# AUTOMOBILE ENGINEER

*Design, Materials, Production Methods, and Works Equipment*

*Editor:* J. B. DUNCAN

*Editorial Staff:* T. K. GARRETT, A.M.I.Mech.E., A.F.R.Ae.S., F. C. SHEFFIELD

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## 1953

IN all probability 1953 will be the most difficult year for the automobile industry since automobiles were first produced in this country. Not even the slump years of the early 1930's posed problems of such complexity and magnitude. How far they can be overcome by the unaided efforts of the industry is a matter of speculation, but there is no doubt that government action could simplify the issue in a considerable degree.

Until 1939 it was considered axiomatic that an industry concerned with quantity production must have a home market that would absorb the major portion of its products. Since the war, this has not, of course, applied to the manufacture of automobiles but it still remains true that continued prosperity in the automobile industry must be based on a large volume of home sales.

The peculiar circumstances of the past five or six years have tended to obscure this fact. In the immediate post-war years there was not very intense competition for export markets. The automobile industry in the U.S.A. was unable to meet its own home demand and had little reason for seeking export markets. The other major foreign producers, France, Germany and Italy, were, for obvious reasons, not in a position to offer serious competition outside their home markets. Nor did these countries choose to copy British austerity with its definite restriction of home sales to encourage the greatest possible volume of export sales.

### *Foreign competition*

To-day conditions are considerably different. American competition has not yet developed in any great degree, apart from one or two markets, but all Continental car manufacturers are now making very strong efforts to sell in markets other than their own. This is occurring at a time when world demand shows a tendency to contract rather than to expand. It is probably true to say that so far as passenger cars are concerned, the demand for essential vehicles has now been satisfied and there will be an increasing tendency for governments to rank passenger cars as consumer goods in or near to the luxury class.

World conditions could be scarcely less favourable. Before the war, the contraction of markets was a commercial hazard that every trading organization had to face from time to time. It was, however, generally possible to make a reasonable forecast of when a trade recession was likely to occur. To-day, the economic disequilibrium between the dollar and other currencies leads to government regulations and controls, which may be imposed or altered at virtually a moment's notice, and which cannot in any way be foreseen.

In this particularly unstable economic climate, the automobile industry has a peculiarly difficult role. For several years the industry has been the leading export industry. In the more difficult circumstances now to be faced, with intensified competition and import restrictions of one sort or another, it may continue to play a leading part in maintaining exports, but only if prices are really competitive.

### *Prices*

Price will certainly be the most important factor during 1953. It is an open secret that at least two new light cars are to be introduced by British manufacturers early this year. There is, however, no reason for thinking that either will make a break with accepted practice. Apart from these two new cars, the British attack on the markets for quantity produced cars must be based on existing models. These models are already well-proven in service, and the question now remains: what can be done to reduce costs and prices.

Without in the least minimizing the importance of progress in design, we would emphasize that for the immediate future the prosperity of the British automobile industry will depend upon production factors, which in turn can be affected by the policy of the Government.

We do not intend to embark upon a discussion of the effect that taxation of company receipts can have on efficiency, although there is much that could be said of the difficulty of replacing capital assets under present conditions of taxation. What must be stressed is that if the automobile industry is to continue its great contribution to export trade, it must be allowed to develop the home market, paradoxical though that may seem.

### *Plant utilization*

Every effort must be made to allow the productive capacity of the industry to be fully utilized. To production engineers this is self-evident, but unfortunately it is not so evident to others. If prices are to be kept to a minimum, production must be maintained at the maximum. This means that an inflexible ratio between home and export sales is no longer feasible. Economic conditions are now so unpredictable that sudden closing of export markets must be expected. If, through import restrictions, exports contract or even cease completely, they may open again

with disconcerting suddenness, and industry must be prepared for this. Meanwhile, what is to happen to the production surplus to the export market? Fresh overseas markets cannot be opened and the only logical outlet is the home market. Unfortunately, it will not be sufficient merely to allow vehicles to be offered for sale on the home market, for here again price is a much more important factor than it was only two years ago. Manufacturers can still do something to reduce prices for the home market, but there is no doubt that the Government could do far more by easing the purchase tax. A precedent for this has already been created by the Government action over purchase tax on textiles. Admittedly, the action in connection with textiles was taken when the industry was already suffering from a high percentage of unemployment. It is to be hoped that in the case of the automobile industry, action will be taken in time to forestall unemployment. Any percentage fall in plant utilization means a much greater percentage increase in unit costs. That this must be avoided if we are to maintain, much less expand, our export markets, hardly needs saying. But it can be avoided only by creating conditions that will allow a healthy home market to develop.

As was said earlier, for the immediate future the production divisions will be more important than the design divisions, since the work of the latter cannot affect the situation for some considerable time. So far as actual manufacturing methods are concerned, there is little that can be done. The production methods for existing models of quantity produced cars have been carefully planned and immense capital sums have been invested in tooling. Probably, the only point on which there is a possibility of appreciable saving through better planning, is in the reduction of investment in work in progress. Here, the automobile manufacturer is in some measure at the mercy of outside suppliers, who in turn are at the mercy of their suppliers. That improvement can be made is however, indisputable.

### Cost reduction

To meet the conditions of 1953, it is essential that every possible means of reducing costs must be considered. It is therefore, pertinent to ask whether the wage structure in the automobile industry is not out of balance with the structures in other industries. Without doubt, the average wage is much higher than the national average, yet the operators are not called upon to apply a higher degree of skill or to work harder than men in other industries. In fact, it is probably no exaggeration to say that the standard of skill required and the actual physical effort expended in most manufacturing processes in the automobile industry are less than those of other industries. Scientific planning and the use of accurate automatic machines have reduced dependence upon operational skill to a minimum, and no other engineering industry has gone so far in introducing mechanical equipment to reduce fatigue.

While we query the wage structure, this is not to say that we wish to see any reduction in the size of the pay packet; on the contrary, we should like to see it increase. What we do want to see is a reduction in unit costs, and to this we believe that a sensible contribution could be made through an overhaul of the wage structure. Since the war there has been much talk of full employment, but too little realization of the amount of hidden under-employment there is in almost every industry, and not least in the automobile industry. If this under-employment can be rectified production would increase and costs would be reduced, without any reduction in total wages.

Decisions of foreign governments are important, but little can be done about them. The decisions of the British Government are also important, and it only remains to hope that they will be such as to help and not hinder. There are also the actions of the industry itself, not only at managerial level but at every level. On the managerial level it is essential that the development of new designs be prosecuted with the utmost vigour as a fairly long term policy, while as an immediate policy every effort must be made still further to employ equipment that will increase productivity and reduce costs. In the final analysis however, the immediate results will depend in the main upon the operators. It is they who must maintain the quality of work produced, and who must make the maximum possible use of all the available production facilities. The management has the duty of supplying the means while the operators equally have the duty of seeing that the best possible use is made of the means at their disposal.

Although the omens may not be propitious, we are convinced that the industry will overcome the difficulties providing there is no restriction on supplies and none on the development of the home market. The major re-organizations carried out in the past few years have brought the production potential to a very high level; new manufacturing techniques have been developed and applied; and at the same time, the planning engineers, ably supported by their Boards of Directors, have shown and are continuing to show great initiative and skill.

## CONTENTS

# THE LANCHESTER LEDA CHASSIS

## *A Vehicle with Laminated Torsion Bar Front Suspension*

THE Leda which is the export version of the Lanchester Fourteen, incorporates pressed steel bodywork for production on a somewhat larger scale than the aluminium bodied version available on the home market. In addition to these two types, a de Ville convertible model on the same chassis is also available.

In designing the new chassis, the intention was to provide the basis for a car of medium size, capable of carrying four persons in comfort. Another requirement was that it should have good road holding characteristics, and adequate performance, to maintain high average speeds with comfort on long journeys. All these factors are of particular importance so far as overseas markets are concerned.

### The engine

The 1,968 c.c. engine develops 60 b.h.p. at 4,200 r.p.m., and as the vehicle dry weight is 3,136 lb, it gives 42·8 b.h.p. per ton. It is interesting to see how favourably this figure compares with that of the post-war Lanchester Ten, in which the b.h.p. ton was 36·4. Not only is the power developed in the new chassis adequate for this class of car, but the improvement has been obtained at a relatively low cost. Considerable production economies have been made possible by the fact that many engine components are common to both this model and the 3-litre Daimler Regency.

In fact, the Lanchester Fourteen engine might almost be described as a four-cylinder version of the six-cylinder Regency power unit inasmuch as the bore and stroke are the same for each. Therefore the pistons, connecting rods, valves, and most of the valve gear are the same for both. Moreover, many of the other components such as oil and water pumps, timing gears, main journal bearings, etc., are also common.

The bore:stroke ratio is 1·42:1 and the connecting rod length:stroke ratio 1·88:1. This gives the fairly high figure of 2,960 ft/min for the mean piston speed at 4,200 r.p.m. at which speed maximum b.h.p. is developed. The maximum b.m.e.p. is 120 lb/in², and the maximum torque is 96 lb-ft, both values being obtained at 1,600 r.p.m. to match the fluid flywheel characteristics. In terms of b.h.p/in² of piston area, the performance is good, the figure being 2·12; the b.h.p./litre is 30·5. When dry the engine weighs 494 lb, so that the b.h.p/lb is 0.12.

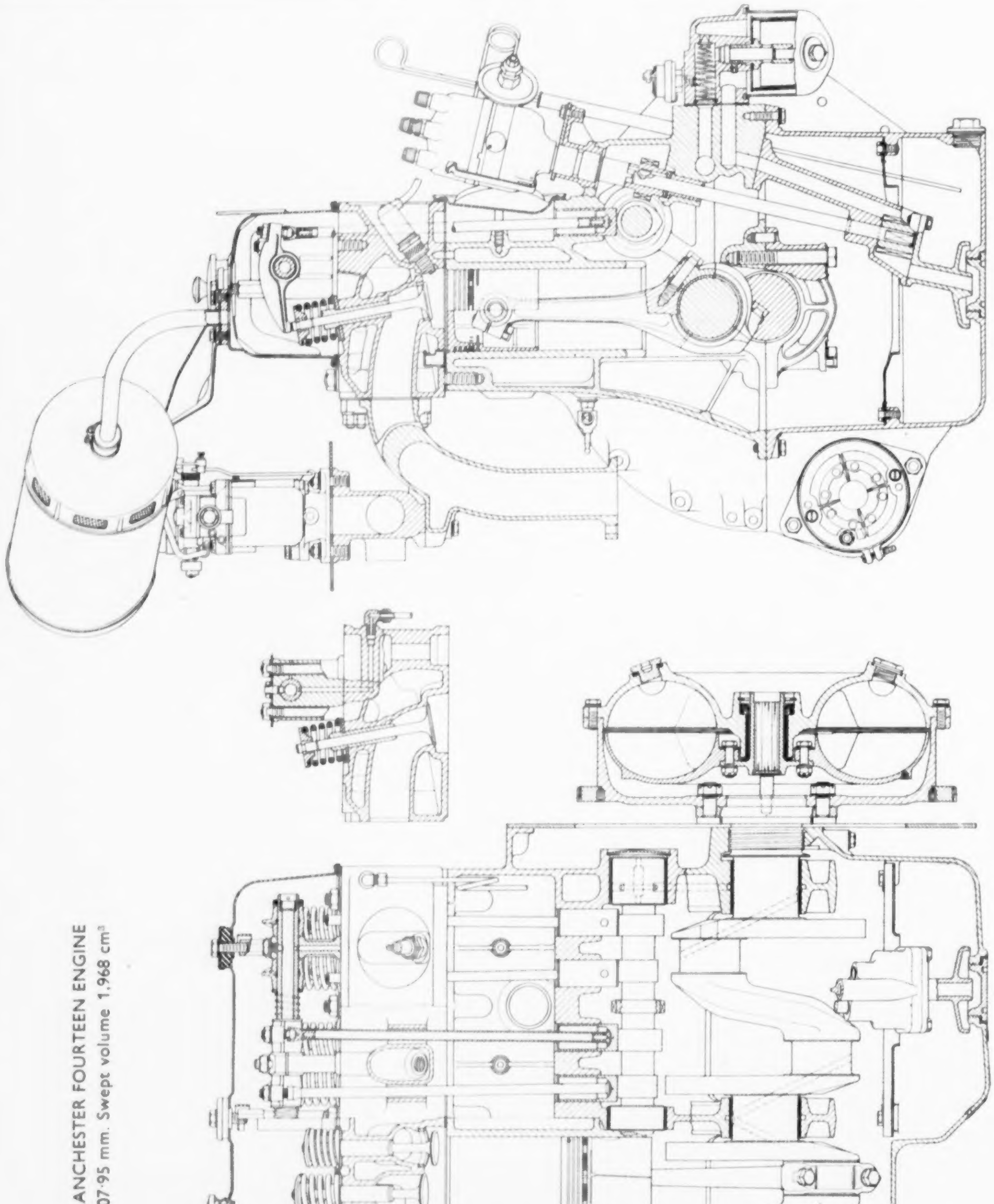
An engine installation angle of 3 deg from the horizontal has been adopted to keep the body floor as free as possible from obstruction. Metalastik mountings are employed throughout. Two solid cylindrical bonded rubber sandwich type units are used at the front. A somewhat unusual feature of these is that three metal discs are incorporated in each, together with two circular rubbers. The central disc is intended to restrain the rubber from bulging unduly. This is necessary because a very soft mix is employed.

At the rear, the sandwich comprises three rectangular plates and two rubber blocks. In this case however, only one unit is fitted. It is positioned vertically, with the centre plate extended upwards and bolted to a bracket on the rear cover

#### SPECIFICATION

*ENGINE: Four cylinders. Bore and stroke 76·2 mm × 107·95 mm. Swept volume 1,968 cm³. Maximum b.h.p. 60 at 4,200 r.p.m. Maximum b.m.e.p. and torque respectively 120 lb/in² and 96 lb-ft at 1,600 r.p.m. Compression ratio 6·7:1. Fully balanced, three bearing forged crankshaft. Overhead valves, push rod operated. Zenith downdraught carburettor with 27 mm choke and third bar.*
*TRANSMISSION: Daimler Fluid Flywheel Gears, pre-selector unit, four forward speeds and one reverse. Ratios: top 1:1, third 1·475:1, second 2·21:1, first 3·84:1, reverse 5·21:1. Propeller shaft, Hardy Spicer open.*
*REAR AXLE: Salisbury semi-floating unit with a hypoid pinion, and a final drive ratio of 4·56:1.*
*FRONT SUSPENSION: Transverse wishbone link system with laminated torsion bar, and Girling DAS 4/35 shock absorbers.*
*REAR SUSPENSION: Semi-elliptic leaf springs with through axle and Girling DAS 6/40 shock absorbers.*
*STEERING: Bishop cam and roller. 3¼ turns from lock to lock. Turning circle 34 ft.*
*BRAKES: Front, Girling hydraulic 2LS. Rear, Girling mechanical, trailing shoe type. Drum diameter 11 in. Shoe width 1¾ in. Total friction area 143 in².*
*TYRES: 6·70 × 15·00 on 4½ in rims. Pressure: front, 24 lb/in², rear, 26 lb/in².*
*DIMENSIONS: Wheelbase 8 ft 8 in. Track: front and rear, 4 ft 4 in. Dry weight 28 cwt. Ground clearance 7 in. Overall length 14 ft 7 in. Overall width 5 ft 5½ in. Overall height 5 ft 3½ in.*


In the Lanchester Leda, a traditional vertical radiator is incorporated with a delightful modern style



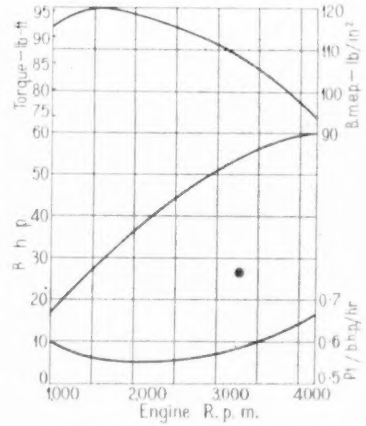
ARRANGEMENT OF THE LANCHESTER FOURTEEN ENGINE
Bore and stroke 76·2 mm × 107·95 mm. Swept volume 1,968 cm$^3$

of the gearbox. The rubber is maintained in compression between the plates, while the weight, and out-of-balance forces of the engine are taken in shear. This compressive pre-load is intended to increase the life of the unit.

Were the pre-load not applied, tensile stresses, complementary to the shear forces, would normally be developed. This would be detrimental since rubber is particularly liable to a form of corrosion fatigue failure when loaded in tension, probably owing to the entry of foreign matter into minute crevices in the material. Under compression, the tendency is to force out these corrosive imporities.

Because of the flexibility of the whole system, it has been necessary to incorporate a torque reaction stop. This is a rubber, approximately semi-ellipsoidal in shape, mounted on a bracket bolted to an adaptor on the left-hand chassis side frame. Under suddenly applied full torque and resonant conditions it bears on the underside of a lug cast on the flange carrying the bellhousing. The bolt holes in the adaptor plate are slotted so that the vertical position of the stop relative to the engine can be adjusted.

Apart from the four-throw crankshaft and three bearing camshaft, the only major difference between this engine and the Regency is that it has an exhaust heated hot-spot instead of a water jacketed inlet manifold. Another difference, of somewhat less importance from the point of view of general layout, is that in the four-cylinder engine, a torsional vibration damper is not employed. This may appear at first sight somewhat incongruous, but it has obviously been omitted from this engine in the interest of economy.




Engine performance curves

The camshaft is carried on the left-hand side of the engine. A Lucas distributor and contact breaker unit is mounted in a boss on the side wall of the crankcase between numbers 3 and 4 cylinders. It is driven in the usual manner by a nearly vertical spindle which has at its lower end the submerged oil pump. Beneath the distributor is the bolted-on oil filter; in front of it is the fuel pump actuated by an eccentric on the camshaft. The dynamo is pivot mounted at the same side and is driven by a conventional triangulated vee-belt arrangement.

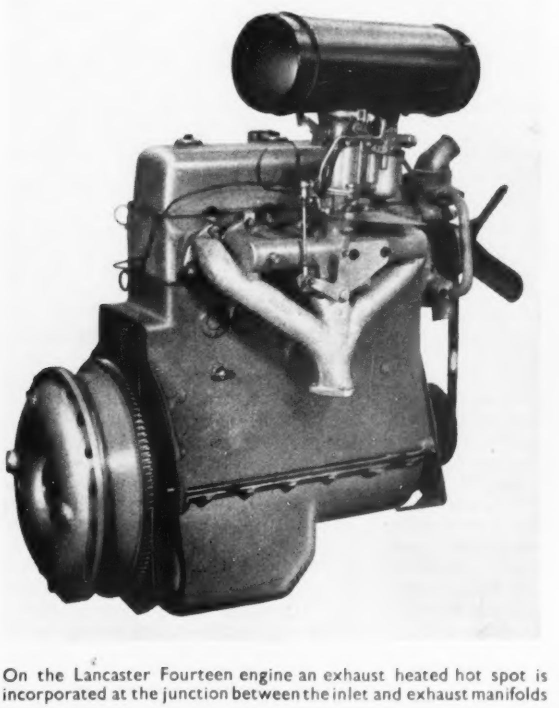
The front end arrangement is much the same as on the Regency, except that a two-bladed fan is fitted instead of a four-blade unit. Above the fan and water pump unit, which is bolted to the front end of the cylinder block, a combined water outlet and thermostat housing is bolted on the end face of the cylinder head. The inlet and exhaust manifolds, and carburettor are on the right-hand side, and on the same side the starter motor is carried low down on the bell housing cover plate. In the Regency the starter was on the left-hand side and rather close to the oil filter, but the right-hand side mounting is possible on the Lanchester because the exhaust pipe layout is different, a single pipe being employed instead of a Y-pipe connection to the manifold.

### Cylinder block and crankcase

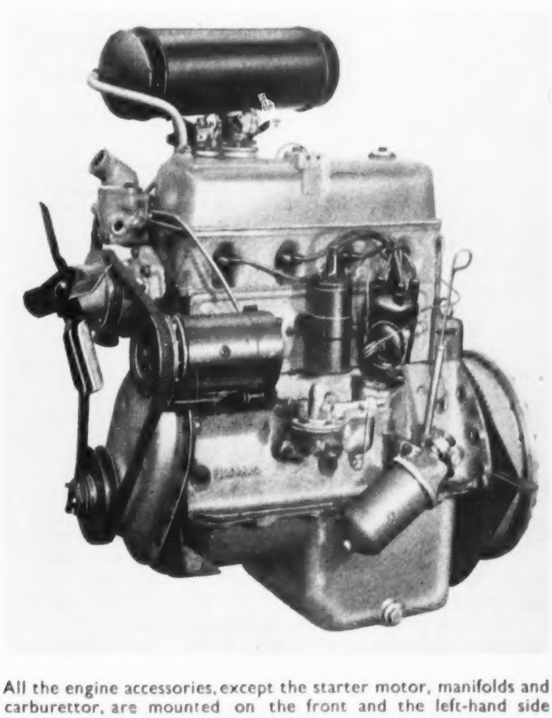
An integral cylinder block and crankcase of cast iron is employed. The cylinder walls are $\frac{1}{4}$ in thick and the bores are hone finished. Generous cooling jackets are incorporated, the minimum width being $\frac{9}{32}$ in between numbers 1 and 2, and between numbers 3 and 4 cylinders. The jacket space at the front and rear is just over $1\frac{1}{4}$ in; between the centre pair of cylinders it is $\frac{7}{8}$ in.

The crankcase walls extend only down to the level of the crankshaft axis. This arrangement allows the greatest possible advantage to be taken of the stiffening effect of the crankcase web and end walls, and at the same time makes it possible to incorporate simple oil sealing arrangements around the crankshaft at both ends.

At the rear, the cast aluminium sump has a semi-circular cut-out machined in it to fit, in conjunction with a similar cut-out in the crankcase, around an oil return scroll cut on the periphery of the tail end of the crankshaft. Around this, the bolted-on bell-housing, together with its $\frac{3}{32}$ in thick closing



On the Lancaster Fourteen engine an exhaust heated hot spot is incorporated at the junction between the inlet and exhaust manifolds



All the engine accessories, except the starter motor, manifolds and carburettor, are mounted on the front and the left-hand side

plate, is bolted on the machined end-face of a rearward extension of the crankcase. At the front of the crankcase, the seal is made by bolting the timing cover, together with the front engine plate, direct to the front face of the cast aluminium sump and the crankcase. The sump end wall is, of course, cast to provide ample clearance around the front journal bearing cap. All joints except that at the clutch bell-housing, are made oil-tight with Walkers Gascoid washers.

Both the camshaft and the main journal bearings are carried in bosses in the crankcase web and end walls. Vandervell bushes are fitted in all three camshaft bearing bosses. The crankshaft journal bearing caps are of cast iron. They are each secured by two ½ in diameter En 25T bolts, and locked by tab washers. Location is effected by two dowels.

The axis of the fully balanced, three-bearing crankshaft is on the same level as the sump face joint

**Crankshaft connecting rods and pistons**

The fully balanced, forged En 15Q crankshaft is carried in three Vandervell D2 bi-metal shell-type bearings. The journal diameter is 2·48 in. The front bearing is 2 in, the rear 2·3 in, and the intermediate bearing 1⅜ in long. In each the diametral clearance around the bearing is from 0·005 in to 0·002 in.

Axial location is effected by two semi-circular Vandervell D2 bi-metal thrust washers located in grooves, one on each side of the front bearing cap. As these washers are carried in the cap instead of in the crankcase bearing boss, the machining operation has been somewhat simplified, and the washers are effective in this position because of the dowel location of the cap.

The overall length of the crankshaft between the front face of the front web and the back face of the rear web, is 14⅞ in. All webs are 11/16 in thick × 4 13/16 in wide, and the inclined crank arms are 1⅜ in thick × 2⅞ in wide. The crank pins are 2¼ in diameter.

In addition to the scroll type oil return at the rear of the crankshaft, there is the usual integral oil thrower ring working in a drainage space between it and the rear journal. Behind the scroll is a flange, to which the fluid flywheel casing is bolted and spigoted. Location is effected by two dowels. At the front of the crankshaft, a 1⅞ in diameter extension carries the timing gears, and a lip-type oil seal supplied by Super Oil Seals and Gaskets Ltd. is employed.

The forged En 15Q connecting rods have a centre-to-centre length of 8 in. They are of H-section, and each has a jet hole drilled in the big end to direct oil mainly on to the thrust face on the cylinder wall. The big ends are split at an angle of 38 deg so that the rods may be withdrawn through the cylinder bores. Two ⅜ in diameter En 25T bolts locked by tab washers secure the caps, and location is effected by dowel tubes around the bolts. Vandervell D2 bi-metal bearing shells, 1 in long, are employed. The diametral clearance for the big end bearings is from zero to 0·0015 in.

A ⅞ in outside diameter, TS14 gudgeon pin is fitted to each rod. Its inside diameter is tapered from ⅝ in at each end to ½ in at the centre. An En 25T 5/16 in diameter pinch bolt clamps the gudgeon pin and engages in a groove in its periphery to provide axial location. In each piston boss the effective bearing length is 13/16 in. Holes are drilled vertically through the boss to assist splash lubrication of the bearing.

Wellworthy 52 alloy, type O.T. pistons are employed. They have split skirts so that the cold clearance may be as small as possible without there being any danger of seizure when the engine is hot. Three plain compression rings are fitted to each piston. Their face width is 0·0615-0·0625 in, radial thickness 0·115-0·122 in, side clearance 0·001-0·003 in, and their gap 0·009-0·014 in. Only one oil control ring is employed. It is positioned above the gudgeon pin. It has the the same gap as the compression rings, but its other dimensions are: face width 0·1865-0·1875 in, radial thickness 0·107-0·114 in, and side clearance 0·0015-0·003 in. All rings are supplied by Wellworthy Ltd.

CAMSHAFT PERFORMANCE DATA AT 5,000 R.P.M.

| | |
|---|---|
| Maximum tappet positive acceleration (flank) | 6,930 ft/sec² |
| Maximum tappet negative acceleration (nose) | 5,070 ft/sec² |
| Maximum tappet velocity | 7·75 ft/sec |
| Lift at cam | 0·235 in |
| Nominal period of cam | 267 deg |

The En 15Q driving sprocket for the timing gear is carried on the front extension of the crankshaft; the drive is transmitted through a Woodruff key. Between this sprocket and the shoulder formed by the front crankshaft bearing is a flanged distance piece. The flange, which is round the rear end of this distance piece, bears against the crankshaft thrust washer. A pressed steel oil thrower ring is clamped between the front of the drive sprocket and the key driven En 8Q fan belt drive pulley. The whole assembly is pulled up by a nut, incorporating the dogs for the starter handle, on the 1¼ in diameter threaded portion on the front end of the crankshaft extension. A tab washer is employed for locking purposes.

The oil seal bears on the boss of the pulley wheel, and is carried in a pressed steel, Z-section circular housing pressed into a flanged hole in the pressed steel timing cover. At the front of this housing, the inwardly turned flange retains the seal, while the outwardly turned flange at the rear forms a lip to prevent oil from running off the wall of the timing cover directly on to the seal. The periphery of the thrower ring is flanged forward to enshroud completely the seal housing.

**Timing gear, camshaft and valve gear**

A Renold and Coventry Chain Co. Ltd. three strand timing chain, of ½ in pitch, transmits the drive to the half speed wheel. Immediately above the driving sprocket, a thimble extends forward from the crankcase wall; it has three radially drilled holes to supply a drip feed of lubricating oil into the meshing point. The centres of the driving and driven sprockets are only 5⅞ in apart, and since their overall diameters are 3⅛ in and 5 5/16 in there is no need for any chain tensioning device.

The En 15R half speed wheel is keyed on to a 1⅛ in diameter extension of the camshaft. It is pulled up against a 1⅝ in diameter shoulder immediately forward of the front camshaft bearing, by a split pinned slotted nut used in conjunction with ⅛ in thick washer on the ½ in diameter threaded end of the camshaft.

The BS4072 thrust plate, ⅛ in thick, registers in the groove formed, round the shoulder, between the front bearing and the boss of the half speed wheel. It is secured to the front wall of the crankcase by two 5/16 in diameter setbolts. There is no provision for fine adjustment to the timing.

All three bearings of the forged 5B.230-2 Keystone brand, camshaft are 2 in diameter. The front one is 1½ in long to take the loads imposed by the drive, the intermediate bearing is ⅞ in long, and the rear one 1¾ in long. Although the length of the rear bearing

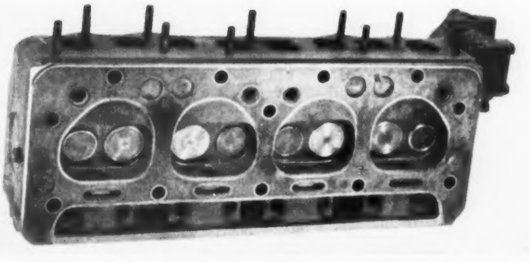
may appear to be excessive, it is probably necessary because the bearing incorporates two flats $\frac{3}{16}$ in wide, which are cut into a large proportion of the periphery. In addition, its oil feed is not quite so positive as that for the other two bearings, since an intermittent feed is taken from these flats to supply the rocker gear.

The camshaft diameter is $1\frac{1}{4}$ in. This is unusually large, and the manufacturers state that a particularly sturdy camshaft is fitted to avoid the possibility that camshaft deflection will spoil the lift characteristics of the carefully designed cam profiles. Cam data is given in the accompanying table. A spiral gear, driving the spindle for the oil pump and distributor, is machined on the shaft between the cams for numbers 3 and 4 cylinders, and an eccentric actuating the fuel pump is similarly incorporated between the cams serving numbers 1 and 2 cylinders.

Piston type tappets of chilled cast iron are employed. They are carried in the crankcase in bosses drilled 1 in diameter. The push rods are made of A4 0·33 per cent carbon steel tube $\frac{3}{8}$ in outside diameter × 20 s.w.g. Each pair of inlet and exhaust push rods is inclined so that the tappets may be positioned close together to leave room for camshaft bearings of adequate length. The angle of inclination in the longitudinal plane is about 1 deg from the vertical. Another inclination of 1 deg is incorporated in the transverse plane in order that the tappets and pushrods shall not come too close to the water jacket. At each end, cyanide hardened En 32A end fittings are pressed into push rods. At the bottom these fittings are spherical faced to seat in the tappets; at the top they are cupped to carry the cyanide hardened En 32A ball ends of the tappet adjusting screws. These screws, also of cyanide hardened En 32A, are carried in the rocker ends and secured in the usual manner by lock nuts.

Forged En 32A rockers are employed. Their bores, $1\frac{1}{8}$ in long, are carburized, and the end pads which seat on the valve stems, are cyanide hardened. They are carried in pairs, one rocker on each side of the En 8Q pedestals which are slightly offset towards the exhaust valve, above each cylinder. This offset is incorporated, of course, in order that symmetrical rockers may be employed despite the difference in size between the inlet and exhaust valves.

Between each pair of rockers the conventional arrangement of a compression spring and two thrust washers around the shaft constrains the rockers against their pedestals. At each end constraint is provided by spring washers together with plain washers retained by split pins through the shaft.



Lozenge-shaped combustion chambers are incorporated in the cylinder head

The rocker shaft is of En 32A, and is carburized. It is $\frac{13}{16}$ in outside diameter × $\frac{1}{2}$ in inside diameter. Pressed steel plugs seal the counterbored ends of the shaft. They are held in position by the same split pins that retain the end washers around the shaft. These pins are passed through the counterbored portion just outboard of the plugs. Had the holes for them been drilled through the plugs, as is done in some designs, time would have been wasted in lining up the holes on assembly.

Four pedestals carry the shaft. Each, together with its cap, is held down by two $\frac{3}{8}$ in diameter studs and nuts. A dowel-ended bolt is screwed through the top of the rear bearing cap to engage in a radial hole in the shaft. This arrangement provides axial location. It also prevents rotation, which, if it occurred, would cause another radial hole, drilled in the shaft for lubrication purposes, to move out of line with the oil feed duct in the rear rocker pedestal. The oil hole and the locating hole are diametrically opposed with their axes in line. This is to simplify the machining process.

All the valves are in line in the cylinder head, but are inclined, with their upper ends towards the ports, at an angle of 10 deg from the vertical. This gives a compact combustion chamber and a relatively straight porting arrangement. Circlips around the upper part of the stems prevent the valves from falling into the cylinder in the event of a valve spring breakage. The exhaust valves, as can be seen from the valve data table, are made of XB steel. This material not only has a high resistance to the corrosive effects of leaded fuel, but also has a low coefficient of expansion by comparison with the austenitic steels used in some designs. Despite this low rate of expansion, it is still necessary to taper the stem towards the head for a distance of about 1 in inside the guide. The exhaust valves are used in conjunction with Bimochrome, Durachrome, or Wellworthy Valmet seats pressed into the cylinder head. The inlet valve seats are cut directly in the head.

Single valve springs are fitted. They are retained between their two seating washers by split, tapered collets. The lower washer is held down on the cylinder head by a collar cast around the upper end of the valve guide. All valve guides are $2\frac{1}{2}$ in long × $\frac{5}{8}$ in outside diameter × 0·3422 in-0·343 in inside diameter.

VALVE DATA

| | Inlet | Exhaust |
|---|---|---|
| Material | Jessop H3 | XB steel |
| Head diameter | $1\frac{1}{2}$ in | $1\frac{5}{16}$ in |
| Throat diameter | $1\frac{13}{32}$ in | $1\frac{1}{4}$ in. |
| Stem diameter | $\frac{11}{32}$ in | $\frac{5}{16}$ in. |
| Seat angle | 30 deg | 30 deg |
| Seats | plain | inserts |
| Spring rate | 340 lb/in | |
| Spring length free | 1·6 in | |
| Spring length installed | 1·469 in | |
| Surge frequency | 23,360 c.p.m. | |
| No. of coils | 4 | |
| Coil diameter | 1·1 in (I.D.) | |
| Wire gauge | 3 s.w.g. | |
| Valve lift | 0·375 in | |
| Rocker ratio | 1·68:1 | |
| Valve crash speed | 4,400 r.p.m. | |
| Valve guide length | $2\frac{1}{2}$ in | |
| Tappet clearance | 0·013 in | |
| Valve opens | 19 deg B.T.D.C. | 54 deg B.B.D.C. |
| Valve closes | 67 deg 54 min A.B.D.C. | 22 deg 54 min A.T.D.C. |
| Ignition timing | 9 deg B.T.D.C. (advanced) | |

### Cylinder head, manifolds and carburettor

The overall dimensions of the cast iron cylinder head are $18\frac{15}{16}$ in long × 4 in deep × $7\frac{1}{4}$ in wide. It is held down by twelve $\frac{7}{16}$ in diameter En 16R studs carried in bosses at the top of the cylinder block. The spark plug bosses are deeply recessed into the left-hand side of the head. On the same side, a machined face forms a seat for the upper edge of the pressed steel tappet chamber cover. A cork washer is employed with this cover in order to allow for any slight misalignment of the head and cylinder block portions of the face and to seal the joint.

The pressed steel rocker cover is held down on top of the cylinder head by three $\frac{3}{8}$ in diameter set-bolts. They are pulled down on a dished retaining washer, under which is a rubber washer $\frac{1}{2}$ in thick × $1\frac{1}{2}$ in diameter, spigoted into a hole punched in the rocker cover. The bolts at each end are screwed into special long nuts, which perform the dual function of holding down the end rocker pedestals as well as the rocker cover. The intermediate bolt is screwed into a special pedestal, the lower end of which is threaded and screwed directly into the cylinder head. This arrangement, together with the steel-reinforced cork washer at the face joint on the head, forms a resilient mounting for the cover, and is intended

to reduce noise. The joint washer is supplied by Coopers Mechanical Joints Ltd.

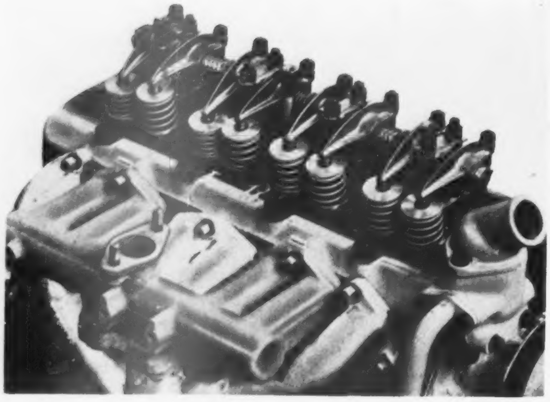
A conventional copper and asbestos gasket is fitted beneath the cylinder head. Compact lozenge-shape combustion chambers are incorporated, and to provide the necessary amount of turbulence for efficient combustion, part of the head on the side opposite the spark plug overhangs the piston at the level of the lower joint face. This gives rise to a squish effect as the piston comes up to top dead centre.

Separate exhaust and siamesed inlet ports are employed. The exhausts are lower down in the head than the inlet ports. This low position, together with an appropriate port contour, ensures that the exhaust valve guide is almost completely buried in the water jacket and is therefore properly cooled. Pressed-in thimbles in the lower face of the head are in line with coolant ducts in the cylinder block. They direct the coolant around the exhaust valve seats.

Although the exhaust port turns sharply from the seat with a raduis of approximately 3/4 in, the angle turned through is only about 65 deg. It is difficult to see how, without increasing the overall height of the engine, and creating cooling problems, sharp bends of this nature can be avoided in exhaust ports; for the exhaust pipe must pass downwards and under the floor, and the manifold has to be well clear of the induction system and below it. The inlet ports turn through approximately the same angle, but with a mean radius of about 1 5/8 in. This restricts the cooling jacket around the valve guide, but it does not matter since the inlet valve, of course, does not require the same degree of cooling as the exhaust valve.

The manifolds are carried on studs, 5/16 in diameter, on the right-hand side of the cylinder head. A cast iron exhaust manifold is employed. It incorporates a cored bracket on which the cast aluminium inlet manifold is bolted. Exhaust gases flow through the jacket to provide heat to a spot below the riser pipe of the manifold. This hot-spot differs from the system incorporated in the Regency, in which heat was supplied by a water jacket. Coopers Cemjo joint washers are fitted between the cylinder head and the flanges of the manifold branches.

A Zenith 46 VIS downdraught carburettor with a 27 mm choke is fitted, together with a Cooper heat shield. The main, slow running, and accelerator jet sizes for the home market are respectively 85, 55 and 50. An 80 main jet is incorporated for export. A 2 mm needle seat is used on all. A balance pipe connects the float chamber with the air intake so that, in the event of the air cleaner becoming clogged, over enrichment of the mixture will not occur. The carburettor is supplied with fuel from a 15 gallon tank by an AC, U.E. type mechanical fuel pump incorporating a gauze filter.



A special nut at each end and a pedestal above the centre of the cylinder head are provided for the three rocker cover holding-down bolts

**Water pump and cooling system**

The water pump is driven at 1·16 times engine speed by a reinforced rubber fan belt supplied by the British Tyre and Rubber Co. Ltd. This belt is 3/4 in wide × 1/2 in deep, and the vee angle is 42 deg. It runs on a cast iron pulley, to the front of which is bolted the two-bladed pressed steel fan. The pulley is keyed on to the front end of the 5/8 in diameter En 56B pump spindle.

A cast iron water pump body is employed. It is closed at the rear by a 1/8 in thick mild steel plate. The spindle is carried in two sealed ball bearings in the nose of the casting. These bearings are separated by a distance tube 7/8 in long. On top of the casting there is a nipple through which grease is supplied to the space between the bearings. To prevent damage to the seals due to pressure built up by over lubrication, an escape hole is drilled axially through the wall of the casting between the two bearings.

A thrower ring is clamped between the inner race of the rear bearing and a collar on the spindle. It works in a drainage space in front of the water seal. The whole assembly, comprising the thrower ring, bearings, distance tube and pulley are pulled up against the collar by a split pinned nut and washer on the front end of the spindle. Axial location is effected at the front bearing outer race, which is held between a shoulder and circlip in the casting. The outer race of the rear bearing has no positive axial location.

Behind the bearing and thrower ring assembly is a spring loaded rubber seal made by the Morgan Crucible Co. Ltd. The front end of the seal is housed in the pump casing, and a moulded-in carbon thrust ring at its rear end bears against the cast iron pump rotor. This rotor is 3 1/16 in diameter and is pressed on to the spindle. Tapped holes are provided for an extractor, in the rear end of the rotor boss.

Coolant circulates from the pump into the cylinder block. Thence it passes up through slots beneath the spark plug bosses, and also through thimble jets pressed into the lower face of the cylinder head. These jets divert the flow round the exhaust valve seats and ports. This system has much to recommend it, for the cylinder walls are better cooled than is the case with designs in which the main circulation is in the cylinder head only, and the walls are cooled by a subsidiary thermosyphon effect. What is perhaps more important is that in the Leda the hot parts of the head are also positively cooled by the flow directed over them. This system is simple and inexpensive.

From the front end of the head the coolant passes through the bolted-on D.T.D.424 thermostat housing and water outlet to the gilled tube radiator. This is supplied by the Coventry Radiator and Presswork Co. Ltd. Its frontal area is 2·78 $ft^2$, and block thickness 1 3/4 in. The thermostat lifts at 173 deg F.

**Oil pump and lubrication system**

The body of the semi-submerged, gear type oil pump is made of cast iron. A drilled extension of the body, forming the oil outlet, is bolted up to the boss on the crankcase wall. This boss also carries the Tecalemit full flow filter. The oil inlet pipe is cast on the bottom cover of the pump. It is drilled out to the same diameter, 1/2 in, as the outlet, and projects downwards into a small well formed by a dish-shaped plug screwed into the base of the sump. Cast integrally around this pipe is an inverted dish-shaped shroud to deflect sludge and heavier solids which might otherwise settle in the well.

The screwed-in plug is incorporated so that the production test-run may be made with an external oil supply connected direct to the pump inlet. In this way any swarf, etc., which may have accidentally got into the engine during manufacture is washed away and removed from the oil by an external centrifuge filter. After the test the plug is screwed in flush with the base of the sump, so that damage will not result if it is struck by any object on the ground. It is peened in position.

Two 1/2 in diameter, En 8Q spindles carry the pump gears, made of the same material. The driven gear spindle is pressed into the pump casing; while the driving spindle is engaged in its gear, and has two flats on its periphery to furnish the drive. Contrary to the arrangement in many other engines, this spindle is not totally enclosed by the pump casing. It is supported at its upper end in a phosphor bronze bush, 1 1/4 in long, in an extension of the boss to which the casing and oil filter are bolted, and at its lower end, for a

length of 1⅛ in, directly in the casing. Both bearings are lubricated by splash, and cup shaped recesses are incorporated around their upper ends to collect the oil and feed it in to the working surfaces.

The top bush is flanged at its upper end to seat in the recess. Above this flange is a hardened steel thrust washer, and above that, the Holroyd Spuncast Holfos bronze, grade JH17, or B.S.S. 2B8 spiral driven gear is keyed on to the spindle. The boss of the gear extends about ¼ in above the end of the spindle, and is slotted to engage the tongued end of the contact breaker and distributor drive.

The sump capacity is 9½ pints. A sheet steel baffle is positioned horizontally about ⅝ in above the oil level. No strainer is employed; and there seems to be little reason for incorporating one, since a full flow filter is in the lubrication circuit and the pump inlet is shrouded. The pump delivers oil from the sump, through the filter, into the ⅞ in diameter gallery drilled in the crankcase wall. An electric oil pressure gauge is screwed into the filter casting, and an oil pressure relief valve lifting at 40 lb/in² is incorporated in the gallery.

From the gallery, three ducts are drilled horizontally, one in each end wall and one in the crankcase web, to connect with three more ducts, one between each camshaft and main journal bearing. Oil passes from an annular groove around each main bearing through two holes in each half bearing shell to the working surfaces. At the front there is a longitudinal hole through which oil feeds from the duct connecting the camshaft and main bearings into the jet tube lubricating the timing drive. Holes drilled in the crankshaft serve the big end bearings, in which jets are drilled to splash oil mainly on to the thrust faces of the cylinder walls. The small ends are splash lubricated.

In the rear camshaft bearing, two radial holes communicate with one another from two chordwise flats on the periphery. The holes and flats are set at such an angle that, as the shaft rotates, they intermittently connect the oil feed with a duct to an external pipe supplying the rocker gear. A longitudinal flat helps to distribute oil over the length of the bearing, and a longitudinal hole through the bearing prevents oil pressure from building up behind the shaft and blowing out the Welch plug that closes the rear end of the bearing boss.

In the cylinder head, the oil passes from the external pipe connection through a lateral passage to a longitudinal one, and thence up vertically through the rear rocker pedestal into the rocker shaft. Radial holes in the shaft communicate with an annular groove around the bore of each rocker. Holes are also drilled in the rocker bosses to pass oil up to a point from which it may run down the arms to the push rod ends.

**Fluid flywheel**

The fluid flywheel has an overall diameter of 13⅞ in and weighs 132 lb. It consists of two main elements. The outer one is the flywheel casing and driving member, and is bolted to a flange on the crankshaft, while the other the driven member, is splined to the gearbox main-shaft. Around the forged En 3A casing is a pressed-on starter ring gear of En 8Q. Grub screws inserted, with their axes parallel to the axis of the ring, at its peripheral junction with the outer member give additional location. Spigoted and bolted to the rear of this outer member is the cast, 195-60 aluminium alloy driving member, the joint being sealed with sodium silicate.

In a 1⅞ in diameter bore at the centre of the driving member is a 2B8 bronze, flanged bush. This carries the hub, on the flanged front end of which is registered and bolted the D.T.D.428 driven member. Bearing on the outer periphery at the rear end of this hub is a Gaco seal housed in the driving member.

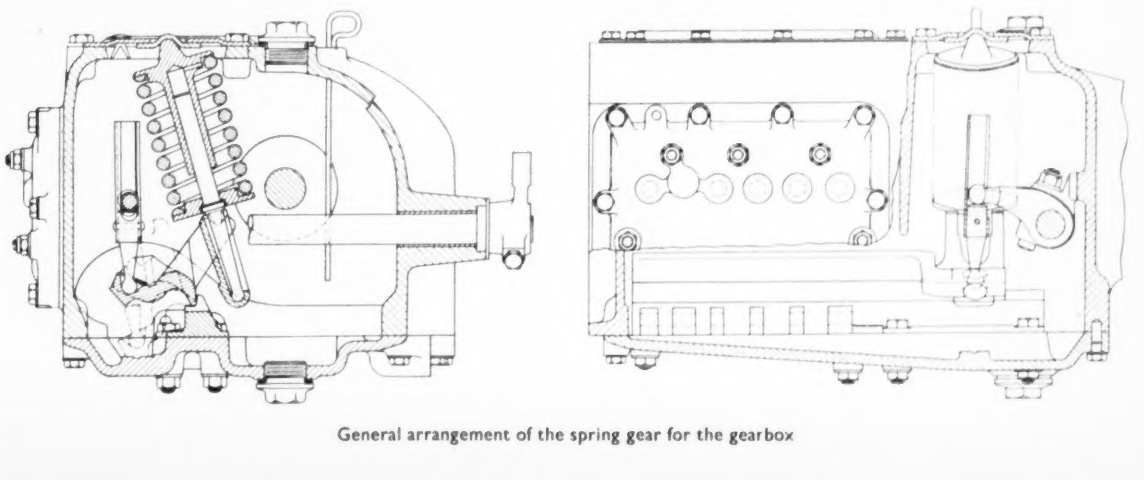
In each of the vortex chambers there are two guide vanes. By this arrangement, adequate directional control is maintained over the flow-vortices. At the same time the mass flow is less restricted than would be the case if these vanes were extended to divide the chambers into a larger number of separate ones. For convenience, two filler plugs are fitted on the rear of the driving member; either may be used for topping up through a hole in the top of the bell housing.

**Gearbox**

The gearbox is a scaled-down version of the Daimler pre-selector unit employed on the Regency, which was described in detail in the March, 1952 issue of *Automobile Engineer*. Therefore only a brief summary of the principles of operation of the unit will be given here. There are two detail differences worthy of mention. The rear extension has been made longer in order to obviate the necessity for an intermediate propeller shaft bearing. This has necessitated casting the extension integrally with the remainder of the gearbox casing, in order to provide a support of adequate rigidity. In the Regency, which has the additional bearing, the rear extension was bolted and spigoted on to the rear end of the gearbox. The other difference is that the centre plate of the rear engine mounting is carried on a cast bracket bolted to an almost horizontal face under the rear extension, instead of to inclined brackets cast on each side of it.

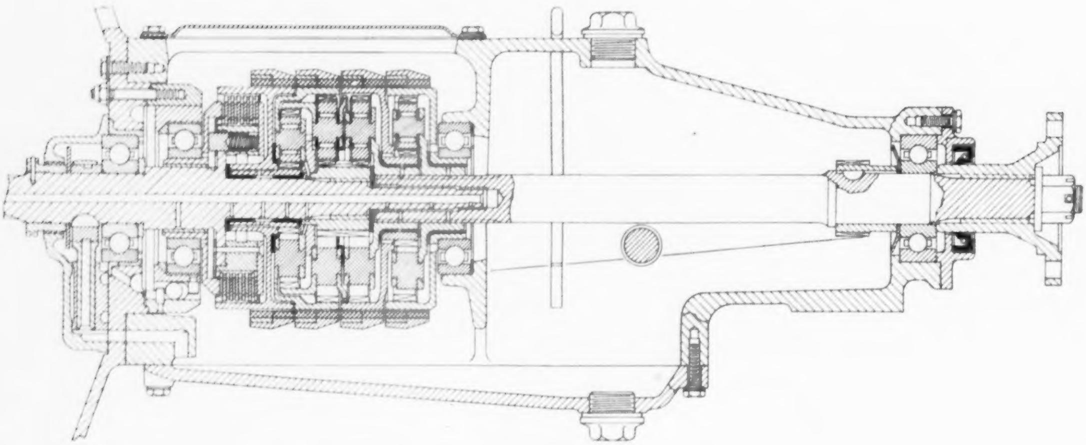
The principal part of the change-speed unit is the running gear. This consists of four simple epicyclic trains, inter-connected to provide various combinations for the different speeds. The appropriate combination is brought into operation by brake bands which furnish the reaction, at the annulus in the case of reverse, first and second speeds, and at the sun wheel for the third speed. Top gear, on the other hand, is engaged by a clutch which locks the whole system together to provide a direct drive. The gear ratios are: top 1:1, third 1·475:1, second 2·21:1, first 3·84:1, reverse 5·21:1.

Each brake band is anchored in such a manner that the torque reactions are equal and opposite. Furthermore, the closing loads used to apply the brakes are also balanced. Thus, the gear trains and the mainshaft are relieved of all asymmetric loading.

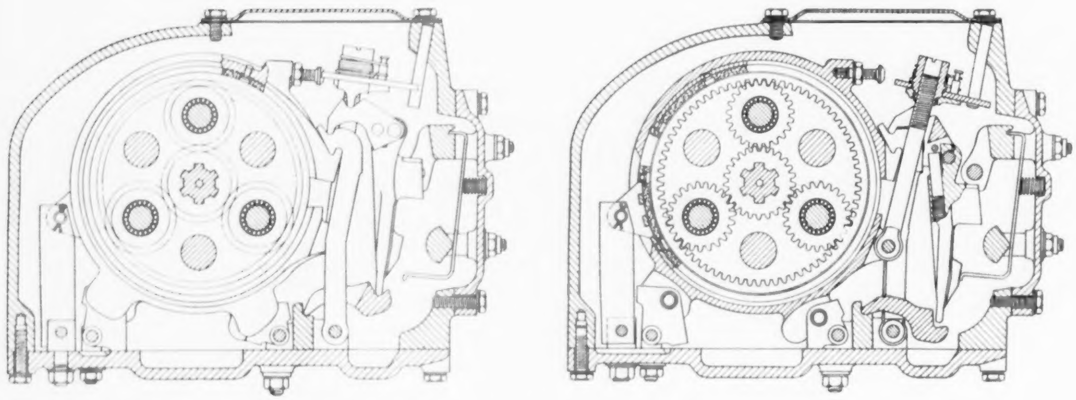


General arrangement of the spring gear for the gearbox

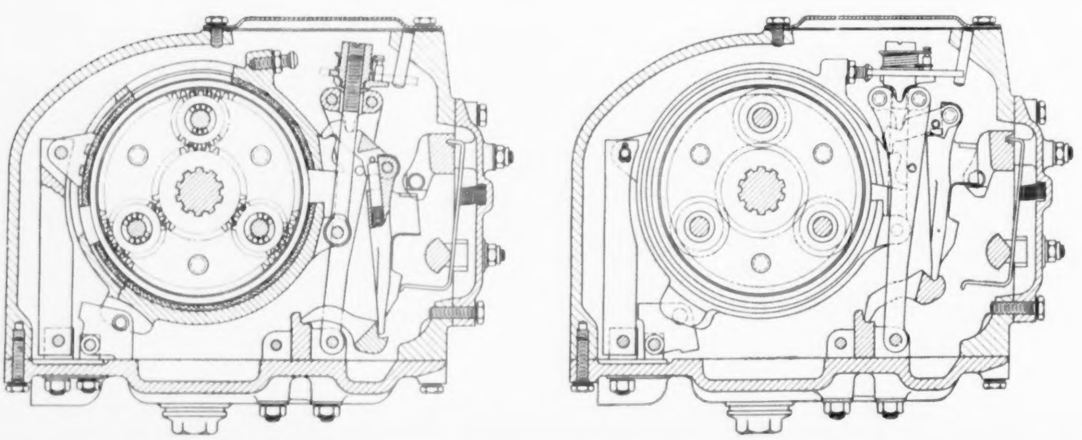
THE PRE-SELECTOR GEARBOX



(a) General arrangement of the running gear



(b) The forward brakes and their operating mechanism



(c) The reverse brakes and their operating mechanism

A bus-bar, extending the whole length of the epicyclic gear and clutch assembly, actuates a series of struts and toggles to apply the closing load to contract the brake bands on their drums and also to engage the clutch for the top speed. Wear of the brake linings is automatically compensated for, as it occurs, by a screw mechanism that shortens the length of the closing tie-rod. In operation a strut, appropriate to the speed selected, is brought into contact with the bus-bar by a striker lever actuated by a camshaft. To control the camshaft position there is a selector lever on the steering column. Its motion is transmitted through a system of bell-crank levers and rods to a cross shaft situated in the box and geared to the camshaft.

### Back axle

A Hardy Spicer, $2\frac{1}{2}$ in diameter, open propeller shaft, having an effective length of $46\frac{3}{4}$ in, transmits the drive to the rear axle. The sliding joint is near the front end of the shaft where it is surrounded by the reinforcing plates at the centre of the cruciform, frame bracing. A suitable proprietary axle, the Salisbury hypoid bevel, semi-floating axle unit, was chosen for this vehicle because it was more economical to use a standard component than to make a special one for the Lanchester car. Moreover, it has been well proved, and little, if any, development work was therefore necessary. The final drive ratio is 4·56:1.

In this unit, all the gears are housed in the one-piece black heart malleable iron casting to BS.310:1947 which forms the axle casing. A pressed steel cover, in which is an adaptor for the oil filler and level plug, is bolted on the rear. A Cooper's Mechanical Joints Ltd., or Flexdid joint washer forms the seal. Hypoid oil of S.A.E.90 viscosity index, conforming to U.S. Specification MIL-L-2105 (formerly 2-105B), is employed, and the capacity is $2\frac{1}{2}$ pints.

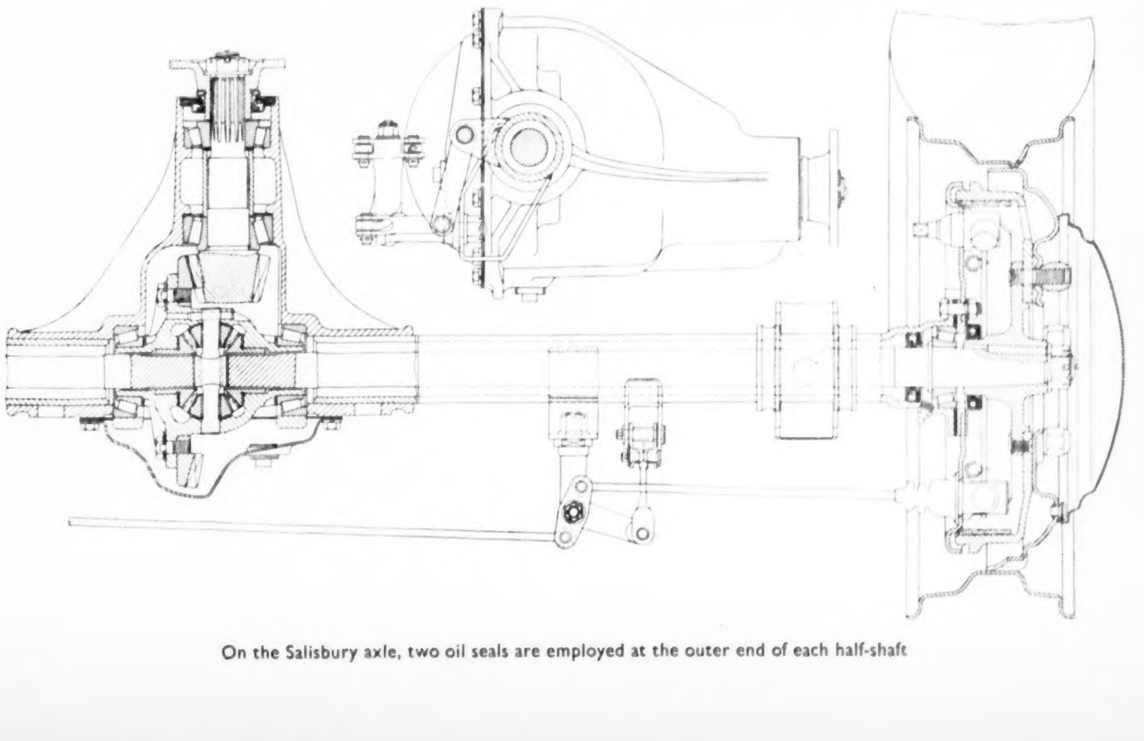
The $2\frac{3}{4}$ in diameter, 9-tooth, hypoid drive pinion is integral with its shaft, and overhung from two taper roller bearings, which are spaced $2\frac{1}{4}$ in apart by a tubular distance piece. The axis of the pinion is spaced $1\frac{3}{8}$ in below the axis of the crown wheel and $\frac{9}{16}$ in to the right of the axis of the differential pinion. At the front bearing, the shaft diameter is $1\frac{1}{8}$ in, while at the rear, it is $1\frac{3}{8}$ in. The inner races, together with their distance tube and the companion flange for the universal joint, are pulled up against the pinion, by a split pinned slotted nut and plain washer on the $\frac{3}{4}$ in diameter threaded end of the shaft. The inner faces of the outer races of the two bearings are each located against a shoulder in the casing. Shims are fitted between the rear race and its shoulder to provide axial adjustment, and the bearing pre-load is regulated by shims between the inner front race and the distance tube.

A thrower disc intended to relieve pressure on the oil seal is clamped between the internally splined boss of the companion flange, and the front race. In front of this, and bearing on the periphery of the boss, is a lip-type oil seal of unusually large diameter. This seal, supplied by Super Oil Seals and Gaskets, is pressed into the front end of the axle casing; its pressed steel housing incorporates a forward turned lip which extends into a U-section, shroud ring, pressed on the boss of the companion flange.

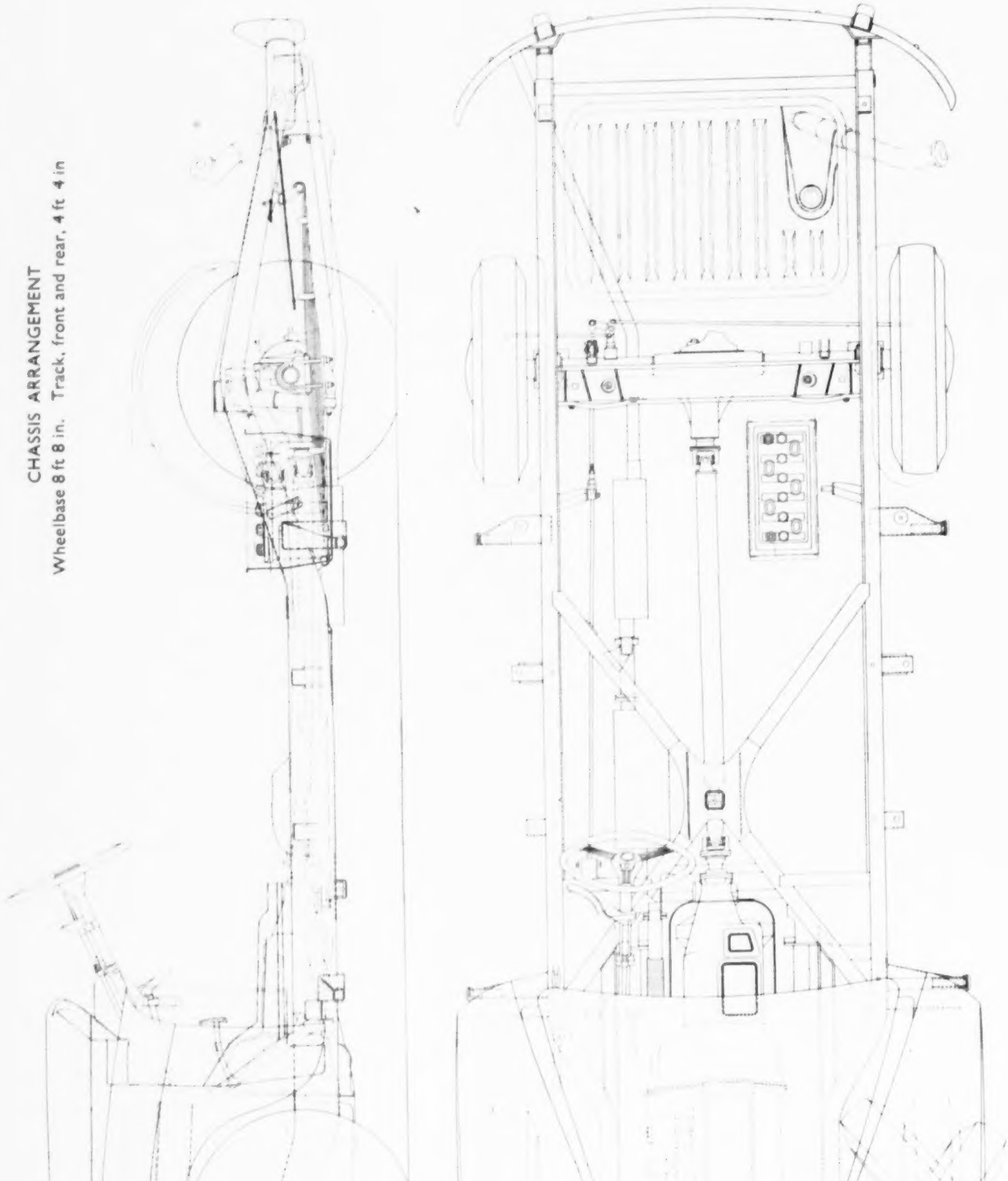
The En 35A crown wheel is $7\frac{3}{4}$ in pitch diameter and has 41 teeth. The component is Lubrised after lapping to facilitate run in. It is drilled and tapped for the eight $\frac{3}{8}$ in diameter set bolts securing it to the inner face of the flange around the one piece black heart malleable cast iron differential cage. Two En 35A differential pinions are employed, and their $\frac{5}{8}$ in diameter En 35A spindle is located by a peg driven into a hole drilled diametrically through it and through the casing. Flats are machined on the spindle to allow oil to pass to both the journal bearing surfaces and the $1\frac{5}{8}$ in diameter En 2A spherical thrust washers. Flat thrust washers, $2\frac{5}{16}$ in diameter, of En 2A, are fitted behind the 2·67 in pitch-diameter En 35A differential pinions, which have their bosses bearing directly in the differential cage. The journal bearing length of the differential wheels is $\frac{11}{16}$ in while that of the pinions is $\frac{3}{4}$ in. The effective length of tooth engagement is $\frac{5}{8}$ in.

Two taper roller bearings, $4\frac{1}{2}$ in apart, are mounted one on each side of the cage, to carry it in the casing. The nip of the bearings, and the mesh of the crown wheel, are regulated by shims between the inner races of the bearings and a shoulder on each side of the cage. An interference fit between the outer races and the shoulders in their housings, provides the axial pre-load. These shoulders are formed by the inner end of the bosses into which the $2\frac{1}{2}$ in diameter by $\frac{7}{32}$ in thick 0·30/0·35 per cent carbon steel axle tubes are pressed and plug welded. Thus, continuous support is provided all round the bearings, which are retained in position by bolted-on bearing caps.

At the inner ends of the $1\frac{1}{8}$ in diameter axle shafts, where they are splined to engage in the differential wheels, the diameter is increased to $1\frac{3}{16}$ in to reduce the weakening effect of the splines. Between these two ends is an En 19 cylindrical thrust block



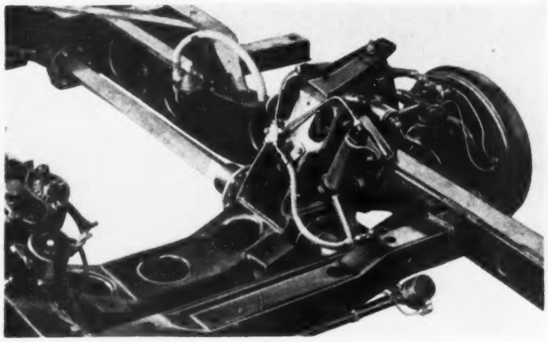
On the Salisbury axle, two oil seals are employed at the outer end of each half-shaft



CHASSIS ARRANGEMENT

Wheelbase 8 ft 8 in. Track, front and rear, 4 ft 4 in

with its axis in line with that of the shafts. A hole $\frac{25}{32}$ in long × $\frac{11}{16}$ in wide is broached in this block so that there is ample clearance between it and the differential pinion spindle which passes through it. The tapered roller hub bearings take thrust in the outward direction only. Inward thrust is transmitted through the axle shafts and cylindrical thrust block to the opposite bearing. The clearance hole in the thrust block prevents any load, due to thrust on the axle shafts, from being transmitted to the differential pinion spindle.



Vernier end-fittings carry the laminated torsion bars of the front suspension

The inner race of each hub bearing is located against a collar integral with the En 19 axle shaft on to which it is assembled from the outer end, and the outer race is housed in the upset-flanged end of the axle tube. This outer race is retained in its housing by a ring bolted to the flange. Shims are fitted between the ring and the flange to set the end float of 0·004 in-0·008 in on the bearing, and to enable further adjustment to be made to compensate for wear.

Two lip-type oil seals, supplied by Super Oil Seals and Gaskets or Retainers, are fitted, one each side of the bearing. The inner one is housed in the axle tube and bears on the axle shaft; the outer one works round the wheel-hub boss, and is in a pressed steel housing. This housing is secured to the brake back plate by the four $\frac{3}{8}$ in diameter bolts attaching it to the bearing retainer ring on to which it is spigoted. To lubricate the bearing, a grease nipple is provided immediately outboard of the seal in the axle tube.

A split pinned slotted nut on the $\frac{7}{8}$ in diameter threaded end of the axle shaft pulls the wheel hub on to a $2\frac{1}{2}$ deg taper. The drive is taken through a plain key. A cast iron brake drum is spigoted on a flange around the hub and secured in the usual manner by countersunk set screws. A standard wheel, made by Rubery, Owen and Co. Ltd., is carried on five $\frac{1}{2}$ in diameter studs screwed into the hub flange. It is secured by conical ended nuts. Large section tyres, 6·70×15·00, are fitted on the $4\frac{1}{2}$ in rims.

**Rear suspension**

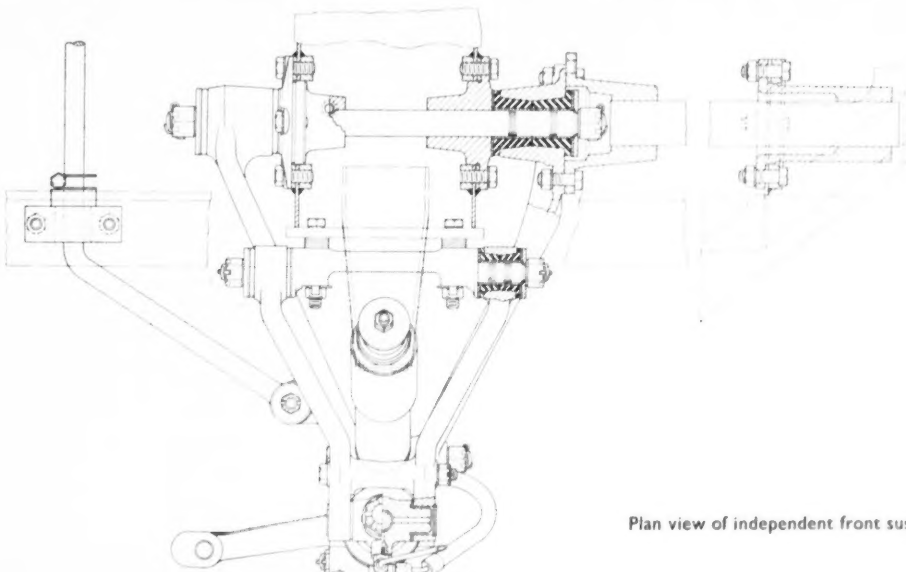
At the rear, the semi-elliptic leaf springs are fastened by $\frac{7}{16}$ in diameter U-bolts to brackets under the axle. They work in conjunction with Girling DAS 6/40 telescopic shock absorbers, in which rebound stops are incorporated. Separate rubber bump stops are mounted beneath the frame side members. The shock absorbers are inclined inwards at the top, at an angle of approximately 20 deg. This was done because it was not possible to install them vertically in the space available. The argument, sometimes put forward, that inclination of the shock absorbers gives an anti-roll effect is fallacious.

In each rear spring there are eleven leaves, $1\frac{3}{4}$ in wide by $\frac{1}{4}$ in thick. The overall length between the spring eye centres is about $22\frac{13}{16}$ in. Four tensioning plates hold the leaves together. Phosphor bronze bushes are fitted in the spring eyes and shackles. They are carried on $\frac{5}{8}$ in diameter, En 32B pins, in U-brackets welded to the underside of the frame. The deflection to the fully laden position is 3 in, and to full bump $6\frac{1}{2}$ in. A periodicity of 75 c.p.m. is obtained with the rear end rate of 70·7 lb/in. The unsprung weight is 332 lb.

**Front suspension**

The front suspension is most interesting because a laminated torsion spring is used. Many advantages are claimed for this type of spring. One of the difficulties associated with the incorporation of a conventional circular-section torsion bar is that owing to its length it is not always easily accommodated. With the laminated torsion bar, however, the length may be varied to suit almost any practical requirements by the simple expedient of employing different numbers of leaves.

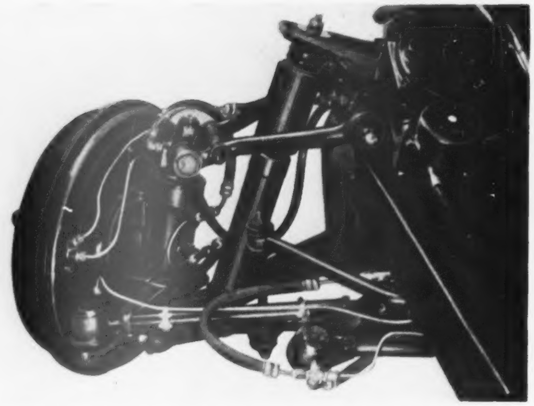
Material is used uneconomically in the conventional bar, because the stress varies more or less uniformly from zero at the axis to the maximum permissible working stress at the periphery. Logically, the first step towards the more economical use of material is to employ a bundle of circular section bars, each of small diameter. In this way, the distance



Plan view of independent front suspension

from axis to periphery is much reduced while the stress at each surface remains the same; therefore the stress gradient is increased considerably, and a greater volume of material stressed to its maximum useful limit. However, the anchorage of the ends calls for relatively expensive manufacturing processes.

The next step is to employ a bundle of square section bars, the end attachment of which is somewhat easier. However, these are not quite so efficient because in a square section, the stress is distributed more or less linearly from zero at the centre to the periphery of a circle of radius $\frac{a}{2}$, where a is the length of the side of the square, inscribed in the square section. It then falls to zero, or nearly so, at the others. Thus, there is a further waste of material in the areas adjacent to the corners.



A view from the front of the double transverse wishbone link I.F.S. system

In the case of the laminated torsion bar, the stress distribution over the section is much the same except that it is a maximum at the periphery of an ellipse whose major and minor axes respectively meet the centres of the short and long sides. It follows that there is an even less efficient use of the material adjacent to the corners than with a square section, but end attachment is simple. A test of its efficiency may be made by comparing the volume of the laminated bar with that of a circular section bar for the same duty.

Unfortunately, it is impossible to generalize on this subject, since the efficiency of the laminated bar naturally increases with the number of leaves employed. There is, however, a limit to this number because of the efficiency falls off and the stress distribution changes radically when the ratio of the lengths of sides, a:b, increases beyond a certain critical value. Moreover, thin leaves have a large ratio of surface area to volume, and are therefore more susceptible to corrosion than thick ones. Because space requirements are more easily met with laminated bars, it might also be considered desirable to obtain greater reliability with the laminated bar by designing for lower stresses than are normally used in the more conventional type. This could also simplify the production process, since shot peening and pre-stressing treatments, often used to improve fatigue characteristics, would no longer be necessary.

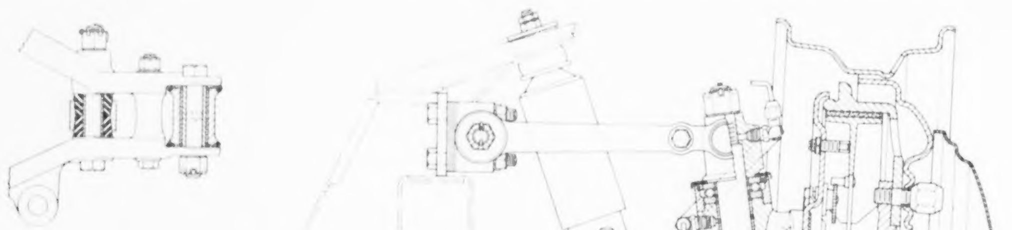
Other advantages are claimed for the new spring. One is that the safety factor is improved because one fractured leaf will not immobilize the vehicle as would a broken circular section torsion bar. There is a certain amount of inherent friction damping in a laminated spring construction but, because of its variable characteristics, this is not universally accepted as a good feature.

There is nothing unconventional about the general layout of the suspension, which is of the double transverse wishbone type. The swivel pin angle is 8 deg, the camber angle 1 deg, and the toe in $\frac{1}{8}$ in. The large section tyres fitted, $6{\cdot}70 \times 15{\cdot}00$ in with $4\frac{1}{2}$ in rims, have a self castoring action, and therefore the castor angle is zero.

Girling DAS 4/35 telescopic shock absorbers are mounted between the lower wishbone link and a bracket overhung from the top of the frame. They incorporate both the bump and rebound stops. The unsprung weight is 97 lb. To the laden position, the angular deflection of the torsion bar is 47·5 deg, and to full bump 13 deg. The spring rate is 260 lb/deg twist of the bar, and this gives a periodicity of 68 c.p.m.

The torsion bar is made of En 45A with a Brinell hardness of 364-444. It is made up from five leaves 0·264 in thick $\times$ 1·32 in wide $\times$ 30·125 in long. At each end, it is carried in a square hole in a flanged cylindrical adaptor. A certain amount of end float of the lamination pack is permitted between the rear end of the lower wishbone spindle and a locating plate welded to the cruciform brace at a position in line with the bar. The flange around the rear adaptor has 18 equally spaced holes, $\frac{21}{32}$ in diameter drilled through it, while that at the front has 20 holes.

At the rear six $\frac{3}{8}$ in diameter bolts are used to secure the adaptor to its support bracket on the frame cruciform member and, at the front, five are



For camber angle adjustment, shims are fitted between the frame and the forging that carries the inner bearings of the upper wishbone

employed to secure it to a flange round the lower rear, wishbone bearing. This is because the number of bolts must be a factor of the number of holes in order that the bolts may always be replaced in the same position in the bracket when adjustment is made to the angular setting of the adaptor.

A vernier adjustment is obtained in the following manner, and it may be readily understood if the fact that the number of bolts used does not affect the adjustment is borne in mind. If the bolts are removed and the rear adaptor is rotated about its axis until the next set of holes in the flange line up with the bolt holes in the bracket, it has turned through an angle of $\frac{360}{18} = 20$ deg. Then if the front adaptor is rotated, in a similar manner, in the same direction it turns through an angle of $\frac{360}{20} = 18$ deg. Thus the rotation of one end of the bar relative to the other is 2 deg which is $\frac{1}{2}$ in in terms of wheel displacement.

The forged En 15S H-section lower wishbone is in two pieces held together by a $\frac{3}{8}$ in diameter bolt passed through drilled bosses whose axes are $1\frac{3}{8}$ in inboard of the outer bearing. Each arm is tapered from $1\frac{3}{4}$ in deep at its inner end to $1\frac{1}{4}$ in deep at the outer end, and both are $\frac{5}{8}$ in wide with $\frac{1}{4}$ in thick webs. A boss formed on each of the two inner ends carries a Metalastik Bonded Cone divided bearing. The axes of the inner bearings are $10\frac{3}{8}$ in from the centre line of the chassis. Both the rubber and the tube around which it is bonded, are flanged at the outer end of each half bearing to cushion the fore and aft loads.

A $\frac{25}{32}$ in diameter En 8Q spindle is common to both bearings. It is carried in two flanged bosses bolted to, and spigoted in, the inner face of each of the two 12 s.w.g. vertical walls forming the front and rear of the frame cross member. The holes in this member, through which the spindle passes, are each exceptionally well reinforced by two welded-on rings. That on the outer face is simply a 12 s.w.g. doubling plate, while the other, $\frac{3}{8}$ in thick, is drilled and tapped to take the bolts securing the bosses. Axial location of the pivot assembly is effected by a $\frac{5}{16}$ in diameter cotter pin in the front boss. The spindle is threaded $\frac{5}{8}$ in diameter at each end, and shouldered to control the compression load applied to the bearing flanges by the split pinned retaining nuts and $\frac{1}{8}$ in thick washers.

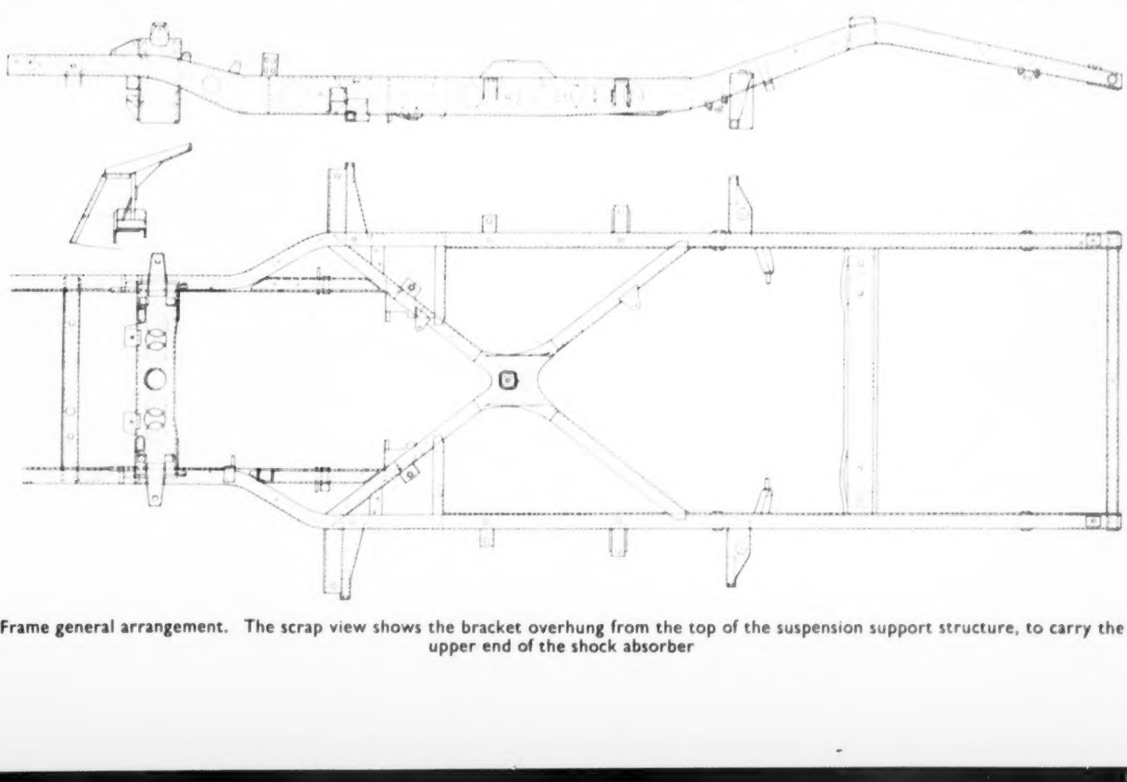
At the outer ends of the two pieces of the wishbone, two lugs are formed to carry an eye on the lower end of the swivel pin. The distance between the axis of the bearing at the outer end and that at the inner end is $13\frac{3}{16}$ in. The eye pivots on a Vandervell, VF2 bronze divided bush around a $\frac{3}{4}$ in diameter En 32B distance piece. The whole assembly, together with two bronze thrust washers between the eye and the lugs, is held by a $\frac{1}{2}$ in diameter En 8Q bolt passed through the wishbone lugs. Lubricant from the automatic system is fed through a radial hole in the swivel pin eye into the space between the halves of the bush. A rubber sealing ring around each thrust washer prevents dust, etc., from entering between the swivel pin boss and the two pieces of the wishbone arm.

Bosses are provided on the wishbone arms, $3\frac{1}{16}$ in inboard of the axis of the outer pin, to take the $\frac{1}{2}$ in diameter bolt which carries the rubber bushed, ring-type fitting at the lower end of the shock absorber. At the upper end, the rubbers of the sandwich-type fitting are clamped between flat retaining washers, one above and one below a 12 s.w.g. cantilever bracket overhung from the upward extension of the frame cross member, on which is mounted the inner bearing spindle for the upper wishbone.

About $4\frac{3}{4}$ in from the axis of the outer pin a boss is formed on the rear arm of the lower wishbone. This boss is drilled, and both ends of the hole are chamfered to take the hemispherical rubber bushes at the lower end of the $\frac{15}{32}$ in diameter drop-link from the anti-roll bar. The eye in the end of the bar is chamfered in a similar manner to seat the two hemispherical rubber bushes at the top of the link. Split pinned slotted nuts on each end of the drop-link, together with four dished retaining washers, one above and one below each pair of rubber bushes, tighten these end assemblies against shoulders on the link.

The anti-roll bar is $\frac{5}{8}$ in diameter $\times$ $37\frac{3}{8}$ in overall length, and the effective or projected, torque arm is $6\frac{13}{16}$ in, its true length on each side being 8 in. Two Metalastik rubber bushed bearings carry the anti-roll bar. They are bolted under the frame side members. The inner end of each bush is flanged and bears against a clip, clamped around the bar by a pinch bolt, to provide the necessary positive axial location.

The forged En 15S upper wishbone link is also in two pieces, held together by a $\frac{3}{8}$ in diameter bolt with its axis $1\frac{3}{8}$ in inboard of that of the outer bearing. Each piece tapers in depth from $\frac{15}{16}$ in at the inner end to $\frac{3}{4}$ in at the outer end. The width is constant at $\frac{5}{8}$ in. Between the axes of the inner bearings of the upper and lower arms, the vertical spacing is $8\frac{7}{16}$ in. At the inner end of the upper link, Metalastik



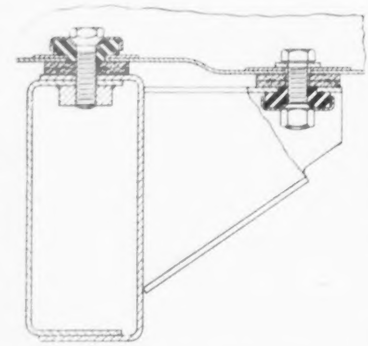
**Frame general arrangement. The scrap view shows the bracket overhung from the top of the suspension support structure, to carry the upper end of the shock absorber**

bushes similar to those used on the lower wishbone, are mounted on ⅝ in diameter pins formed integrally at each end of an En 8Q forging. The axes of these pins are 14 13/16 in from the centre line of the chassis.

This forging is secured by four ⅜ in diameter bolts to a 5/16 in thick plate welded vertically to the outer face of the upstanding end of the frame cross member. Two holes are punched in the inner face of the cross member, so that a box spanner may be used on the bolt heads. Shims to adjust the camber angle are interposed between the pivot pin forging and the plate. In each upper and lower wishbone bearing, the pin has two ¼ in wide knurled bands around it. Since the knurling is rolled on, the ridges are raised above the level of the periphery, and prevent rotation of the pressed-on, flanged inner tubes of the Metalastik bushes.

The two arms of a forged En 32 trunnion are supported in bosses at the outer end of the upper wishbone. This trunnion is carried round the top end of the swivel pin. About 3/16 in below the level of the axis of the trunnion arms, the swivel pin is shouldered and threaded, ⅝ in diameter, for a split pinned slotted nut securing the trunnion. There is a clearance round this threaded portion so that lubricant from the automatic system may pass through a radial hole into the trunnion, round the clearance and into axial holes in the ¾ in diameter arms. The outer ends of the trunnion bosses on the wishbone are sealed by peened-in steel discs so that the lubricant is fed to the Vandervell VF2 bronze bushes pressed on the trunnion arms. Thrust washers of the same material are interposed between the bosses and the shouldered body of the trunnion.

The 13/16 in diameter En 32 swivel pin is carried in two Vandervell VF2 bronze bushes in a knuckle forged integrally with the En 8Q stub axle. Both bushes are 1 3/16 in long; they are spaced 1¾ in apart by shoulders in the knuckle. Round the lower end of the swivel pin is a rubber sealing ring carried in a recess in the knuckle and bearing against a collar integral with the pin. Thrust due to the wheel reaction, is taken from the upper end of the knuckle into a ball thrust bearing. Thence it is taken through the trunnion and the



A typical body mounting arrangement

nut to the swivel pin. A separate collar seated round the shouldered end of the knuckle protects the thrust bearing, and forms a housing for a felt sealing ring held in position between the collar and the upper race by a dished retaining washer spigoted on to the lower face of the trunnion.

Four ⅜ in diameter bolts secure the brake back plate, which is spigoted on to a flange around the stub axle. The En 8Q wheel hub is carried on two taper roller bearings, spaced 1¾ in apart, the inner one being on the 1⅜ in portion and the outer one on the ⅞ in diameter portion of the stub axle. An integral collar in the hub locates the outer race of the smaller bearing, which is assembled into the hub from the outer end. The outer race of the larger bearing is located against another shoulder in the hub and is assembled from the other end.

Also housed in the inner end is a Superfect 268 2 N lip-type seal, which bears round a distance ring separating the inner race and the root flange of the stub axle. Around the seal, a lip turned out at the end of the hub acts as a thrower, so that any grease escaping past the seal is thrown into a trap formed by a Z-section ring secured by countersunk set screws to the brake back plate. A hole is drilled through the brake back plate at the bottom of the trap to allow the grease to escape to the outside of the wheel.

The cast iron drum is spigoted on and secured by countersunk set screws to a flange around the hub. Five conical seating nuts on ½ in diameter studs mounted in the flange retain the wheel. The whole assembly, comprising the wheel, brake drum, hub and its two bearings, is pulled up, together with a 1½ in diameter × ⅛ in thick washer, by a split pinned slotted nut on the ¾ in diameter threaded end of the stub axle. A pressed steel cap is screwed on to the outer end of the hub to retain the grease and prevent foreign matter from entering the bearings.

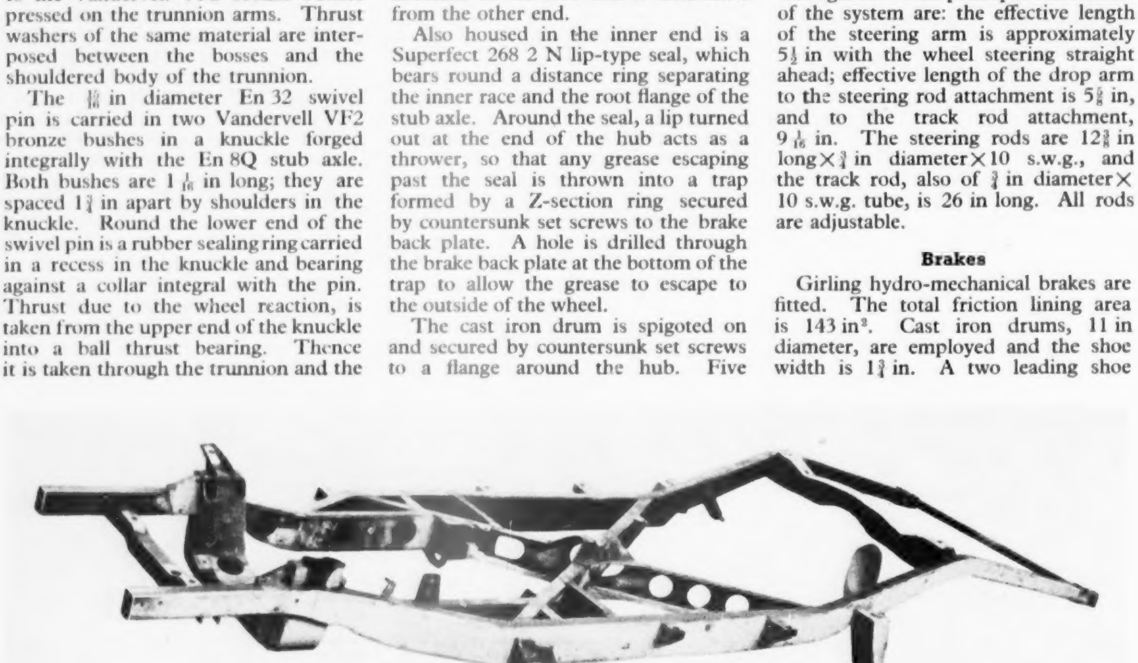
**Steering**

A Bishop cam and roller steering gear is mounted approximately 2½ in behind the front frame cross member. To turn the front wheels over their full range 3¾ revolutions of the steering wheel are required. The front wheel movement is from 36 deg on one lock to 40 deg on the other. The turning circle is 34 ft.

A simple three-rod steering linkage is employed in conjunction with forward extended En 15R steering arms secured on the hubs by two ⅜ in diameter En 160 bolts. The steering rod on one side is connected to the En 15R drop arm; that on the other side is attached to a similar arm operating as an idler lever. The two arms are connected by the track rod which is secured to their forward ends. To obtain the Ackerman effect, the arms are curved so that the track rod joints are offset outboard of the steering rod joints. This would appear to be a simpler and stiffer arrangement than the semi-bell-crank levers sometimes used. Adjustable ball joints are employed throughout. The principal dimensions of the system are: the effective length of the steering arm is approximately 5¼ in with the wheel steering straight ahead; effective length of the drop arm to the steering rod attachment is 5⅝ in, and to the track rod attachment, 9 1/16 in. The steering rods are 12⅞ in long × ¾ in diameter × 10 s.w.g., and the track rod, also of ¾ in diameter × 10 s.w.g. tube, is 26 in long. All rods are adjustable.

**Brakes**

Girling hydro-mechanical brakes are fitted. The total friction lining area is 143 $in^2$. Cast iron drums, 11 in diameter, are employed and the shoe width is 1¾ in. A two leading shoe



The cross member supporting the front suspension is cranked to pass under the engine

layout has been adopted at the front, and two trailing shoes at the rear.

A pistol grip hand brake is employed. Its motion is transmitted through a Bowden cable to the outer end of an 8 in lever, which has its inner end pivoted on a bracket on the frame cruciform bracing. The lever moves in a horizontal plane, and pinned to it, at a point 2⅜ in from the pivot, is a rearward extended link. This layout gives the hand brake a lever ratio of 3·75:1. The link is about 3⅛ in long. Its rear end fitting is slotted where it is pinned to a drop lever at a point 10¾ in below the pivot axis. The reason for incorporating the slot in the end fitting of this link is, of course, to allow the foot brake to be applied without moving the hand brake control.

An adjustable tie rod, operating the rear brakes, is pinned about 1⅞ in above the attachment of the hand brake links. The piston rod from the hydraulic cylinder is attached approximately 4 in below the drop lever pivot. An adjustable tie rod, from a point 3½ in above the pedal pivot, is screwed into the closed end of the hydraulic cylinder. The effective length of the lever arm on which the pedal is mounted is 14 in so that the ratio is 4:1. This gives a brake pedal travel of about 6 in. Compo or Oilite bushers are used at the pedal pivot, and the pedal stem is secured by pinch bolts in a boss on the upper end of the lever.

When the foot brake is applied, the hydraulic cylinder is moved forward relative to its piston to generate the hydraulic pressure to operate the front brakes. At the same time, the piston, which is attached to the drop lever, actuates the mechanical linkage to apply the rear brakes. This linkage is arranged as follows. The adjustable rod from the drop lever which is immediately behind the hydraulic cylinder is connected to a second drop lever pivoted to a bracket 7¼ in behind the spring eyes welded to the inner face of the frame side member. Another adjustable tie connects this lever with a third drop lever pivoted on the axle tube. This, in turn, is connected to a pivoted swinging link compensator, which is also welded to the rear axle tube.

### Frame

All the main members of the frame are of 12 s.w.g., En 2B. The side members are of box section, 2¾ in wide × 5 in deep × ⅛ in thick. This box section is built up from two channel section pieces welded one inside the other, so that their top and bottom walls, formed by the flanges of the sections, are double thickness. This is a sound method of construction, because the tensile and compressive loads due to bending, are much more severe than the shear loads carried by the vertical walls of the section.

Over the portions of the side members which are swept inwards to clear the front wheels when on lock, the box section is discontinued. This discontinuity is effected in the following way. About 16 in behind the point where the front of the cruciform bracing is welded to the side members, a short channel section, lateral member connects each side member with the adjacent cruciform member. Only the outer channel of the side member is continued in front of this. Forward of the point where the front of the cruciform bracing is attached, this channel is swept inwards approximately 6½ in, and then continued straight forward again. This straight portion is boxed in, like the rear, by a second channel section welded inside it. The inner channel is extended straight to the rear until it meets the front of the cruciform bracing to which it is welded. A reinforcing plate is welded to the top and bottom flanges of this channel section to the rear of the point where it is separated from the main box section. Lightening holes are incorporated extensively along the length of the inner wall of each side member, as well as in the cruciform bracing and front cross member.

The cruciform bracing is of channel section, approximately the same depth as the side members. It is increased in depth at the centre to clear the propeller shaft which passes through it. Above and below the centre is a 12 s.w.g. gusset plate. A channel section cross member carrying the rear engine mounting, is bolted at each end to brackets on the two front arms of the cruciform adjacent to the point of attachment of the rear extension of the front side member. Three main cross members are employed. That at the extreme rear is a tube, 1½ in diameter by 12 s.w.g. Above the rear axle, the second cross member is formed by an inverted top hat section, 4¼ in deep × 4 in wide. This member carries the upper end fittings of the shock absorber. It is also drilled for two body holding down bolts.

With the torsion bar front suspension it is not necessary to incorporate at the front such a sturdy cross member as is required to support a coil spring unit. In the Leda, the cross member which is cranked to pass under the engine, is a box section approximately 5½ in wide × 3 in deep at the centre. It is formed by a U-section, closed at the top by a welded on plate. There are lightening holes in both the top and bottom faces of the section. At each side, it is swept up and welded to the frame side members; above this it is extended for about 5 in to carry the upper wishbone bearings and shock absorber attachment brackets. Two brackets, approximately 12 in apart, and equally disposed about the longitudinal centre line of the vehicle, are welded to the front face of this member to carry the front engine mountings. Immediately in front of this cross member, and gusseted to it, is a bracket welded to the inner face of each frame side member. This bracket carries the steering box. About 9½ in forward of the front face of the suspension cross member there is a subsidiary cross member supporting the radiator. It is a channel section 2 in wide × 2 in deep.

Four brackets on each side, overhung from the frame side members, carry the main body mountings. They are positioned one under the front pillar, one adjacent to the rear pillar, and one in front of and another behind the centre pillar. The front and rear brackets are reinforced to take the Bevelift jack. There are subsidiary body mounting points adjacent to each of the front three brackets on the frame side members. Other mountings are positioned as follows: one above the centre of the cruciform member; one on each side of the cross member over the back axle; and one on each side member at the extreme rear.

The details of these mountings are somewhat unusual. All the outer ones on the front three brackets overhung from the side members have rubber washers, ½ in thick, pulled up under the brackets by a ⅜ in diameter body holding down bolt. The rubbers are spigoted into a ¾ in diameter hole in the bracket, through which the bolt is passed. Interposed between the top of the bracket and the body floor is a 9/32 in thick Balata washer. This arrangement is also used for the mountings on the cross member over the rear axle. At all other points the rubber is above, and spigoted into the floor. The ¾ in diameter hole in the floor is reinforced by a steel washer spot welded on, and the Balata is interposed between the floor and the frame. In all cases the rubber is retained by a dished steel washer with a dimpled hole. Normally a conical ended nut seats in the dimple. Where, because of inaccessibility, it is impossible to fit a nut, a conical seat is formed under the head of the bolt which is screwed into a slug welded in position.

### Electrical equipment

Lucas 12 volt electrical equipment is used throughout. A GTW11A battery, of 64 amp hr capacity and a 10 hr rate, is employed. It is served by a C 39 PV-2 dynamo operating in conjunction with an RB 106 regulator and cut-out. The system is protected by a SF6 fuse unit. An M418G starter motor with a 10 tooth pinion engages 113 teeth on the flywheel ring gear. For the first thousand models produced, a DVX 4A contact breaker and distribbutor unit was used, but this has now been changed to a DM2 unit. Both have a vacuum advance and retard control and in each case supply is from a B12 coil. Lodge CLN plugs are fitted. Other electrical equipment includes F700 head lamps, 488 side lamps, 464 tail and stop lamps, wind tone WT 614 horns and SF80 trafficators.

### Chassis lubrication

A 21-point Thermal automatic chassis lubrication system is employed. The only components lubricated by hand are the propeller shaft joint, wheel hubs, steering box and water pump The principle of operation is similar to that of the system on the Regency.

# MACHINE TOOL CONTROL

## *An Interesting Use of a Hilger Projector and Scale*

THERE are, from time to time, occasions when it is desirable to convert standard machines into special-purpose machines by means of special attachments. An example of this practice is shown in the accompanying illustration of a Webster and Bennett 48 in boring mill with a Hilger projector and scale to facilitate machining the annular grooves in the bore of a gas turbine compressor casing. This adaptation not only simplifies the production of work to a high degree of accuracy but also increases the rate of output.

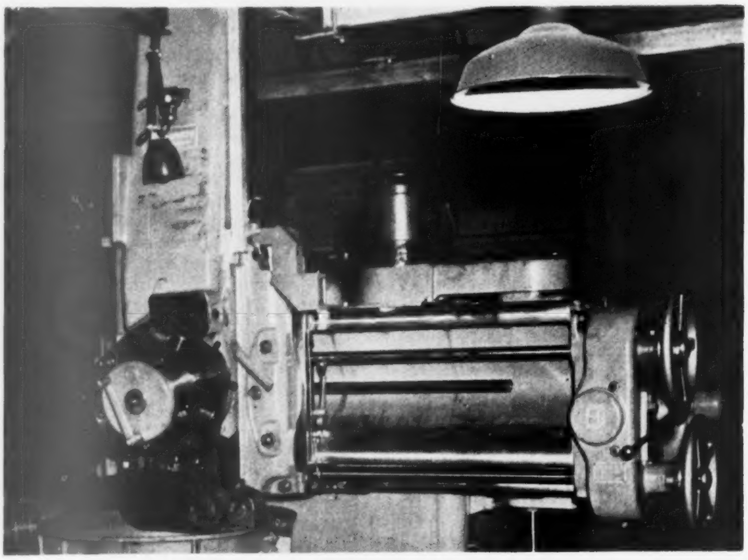
Extremely accurate machining is necessary on the component. The widths, depth and pitching of the grooves must be maintained to very fine tolerances, which are too close to be held easily in quantity production by the use of stops and graduated dials normally provided on a machine tool. On this adapted machine the tools are accurately located by means of optical equipment, and the use of stops and graduated dials is completely eliminated.

In addition, although the cutting tools are in an obscured position, the operator can control their movements precisely and accurately repeat the position of the cutter. By reference to the projection screen, the operator can control, to fine limits, the machining of the series of grooves without any other means of measurement or inspection.

The aluminium alloy turbine casing is in halves, bolted together, with grooves in the bore that vary in width but are of the same form and diameter. These grooves are disposed axially in groups of three and two. Grooves in the same group are the same width, but the width and pitch increase in successive groups. There is also a single wide groove at one end of the casing. The actual machining operation includes facing the end to give the correct casing length, rough and finish machining the grooves, and rough and finish forming undercuts to provide a "tee" form.

This would be a long and tedious operation if normal machine shop methods were employed and there would be considerable risk of scrapping the component. The new arrangement on the boring mill has shown a saving of 82 per cent over the old method. Furthermore, the old method was also dangerous, since the operator had to look continuously at the cut not only for setting but also when the actual machining was in progress.

The optical equipment includes a special scale fitted to the side of the turret slide, and a ×25 magnification surface projector with a screen and a 12-volt lighting unit that is fitted to the saddle of the machine. Through the use of these two units the turret slide, and consequently the cutting tools, can be set with great precision. The scale is a 1½ in section bar of material similar to that of the work, and therefore having the same coefficient of expansion. Temperature changes in the shop will not affect the co-ordination of the set-up.



Webster and Bennett boring mill with Hilger projector and scale

There are two sets of graduations. One is for the preliminary positioning of the turret slide without reference to the projection screen. The other is viewed through the surface projection screen for precise setting. This second set of graduations is engraved on the lapped ends of a series of hardened rollers, which are set in the bar. Surface projection on the end of the rollers is seen on the screen.

The scale is mounted in two dovetail slots, one at each end. The upper slot has an adjusting screw for locating the scale accurately endwise relative to the bottom face of the casing which is located in the fixture. When this adjustment has been made the scale is clamped at the upper end. At the lower slot the scale is free to move endwise to permit expansion or contraction caused by changes in the shop temperature.

It will be appreciated that accuracy in grinding the cutting tool width is also important. Composite tools are used so that after regrinding, shims can be inserted between the halves of the tool and the whole is then bolted together. The tool has an accurately ground shank and is clamped in a tool holder with a precision cut slot and the necessary locating stops. The tool holder is bolted on the end face of the bracket type casting that can be seen mounted on one face of the revolving turret of the machine. This development has been brought about by co-operation between Alfred Herbert Ltd., Hilger and Watts Ltd. and the organization doing the actual machining.

## Radio Interference

BY a regulation made on November 24, 1952, and laid before Parliament on November 27, 1952, the Postmaster General is empowered to ensure that from July 1, 1953, suppression devices are fitted to the ignition systems of motor vehicles, motor cycles, vessels or engines manufactured, assembled or imported by any person in the United Kingdom in the course of business and to be sold or let on hire, or offered for sale or for hire. Vehicles or engines intended for export are exempted from the Order. For existing vehicles no compulsory action is envisaged at present and owners are expected to collaborate by voluntarily fitting a suppressor.

The technical requirements, and suitable apparatus for measuring and testing, are fully detailed in Order No. 2023, *The Wireless Telegraphy (Control of Interference from Ignition Apparatus) Regulations* 1952, published by the Stationery Office, price 6d.

# STROBOSCOPIC ANALYSIS

## *Developments in Measuring and Testing Methods for High Speed Machinery*

THE development of high speed mechanisms inevitably created a demand for means by which the mechanisms could be investigated, studied and analysed during actual operation. Dynamic problems became increasingly important, since in most machines, centrifugal and inertia forces are super-imposed upon the basic strains within the structures. Almost invariably, oscillating systems are set up in the elements of high speed mechanisms, either through the elastic properties and masses of the elements or the masses by which they are loaded, and generally damping is effected insufficiently or not at all.

A combination of external forces with the oscillating system produces motions that could not be foreseen and therefore could not be considered when the machine or mechanism was designed. These oscillatory or vibratory motions do not develop until high speed is reached, and they cannot be analysed while the mechanism is stationary or moving only at slow speed. A stroboscope is indispensable for the study of such periodic motions.

For many years, the only stroboscopes available were of the slit disc or slotted drum type. For simple observation of periodic or cyclic movements and for measuring rotary speeds, they are still widely used to-day. Essentially, these instruments function by obstructing the view of the observer for the greater part of the cyclic movement, the view being cleared for only a fractional part of the movement and at a frequency substantially the same as that of the movement under observation.

### Disc-type instruments

When a rotary movement is observed through the slit in the rotating disc of a stroboscope, the object is seen during only a small fraction of the whole revolution, and always in the same phase position of the moving mechanism. This gives the impression that the object is standing still. If the speed of the slotted disc is changed to a value a little lower or a little higher, the cycle of movement will appear as a slow backward movement in the first case and a slow forward movement in the second.

Theoretically, it would be possible to use this simple type of stroboscope to observe the most rapid motions; in practice, however, an upper limit is set by the fact that at high speeds not enough light from the observed object reaches the eye to form a clearly discernible image. For example, if a disc rotating at high speed is to be viewed through a split-disc stroboscope, in order to get a sharp image, the slit must be so narrow that the view is cleared for only one-third of an angular degree. For the duration of the remaining $359\frac{2}{3}$ degrees, the view is obstructed. This means that only about 1/1000th of the light reflected from the object reaches the eye. Therefore, if the object is to be distinctly seen, it must be intensely illuminated.

### Illumination intensity

An illumination intensity of 100 to 1000 lux reflected from the object is necessary for normal comfortable observation. To get an equally bright image when the object is viewed through a slit of one-third angular degree in a stroboscopic disc, the illumination would have to be increased one thousandfold, that is, it would need to be from about 100,000 to 1,000,000 lux. To demonstrate the great magnitude of this degree of brightness, it may be mentioned that direct sunlight under the most favourable conditions produces in our latitudes only a brightness of 50,000 lux. It is all but impossible to obtain such extremely bright illumination by ordinary means and without undue outlay.

There are, however, many applications for which a sharply defined image is not necessary. For such, the viewing slit can be widened to cover up to three angular degrees. With these wider slots, daylight and normal illumination will give blurred but clearly visible images. Instruments of this type can be used to determine rotary speeds of motors, engines and aircraft propellers by measuring the speed of the stroboscopic disc or drum at which the object appears to be stationary. A stroboscope so used is really a kind of tachometer for measuring the speed of a mechanism from a distance.

For more subtle investigations, such as testing motors for quiet running and studying valve spring surges in internal combustion engines, these simple stroboscopes are not satisfactory. The field of application was widened when the light-flash stroboscope was introduced about three decades ago. They incorporated neon glow lamps, and they have been gradually developed to a remarkable degree. The underlying principle is the electrical emission of intermittent light flashes of relatively short duration, which are controlled so that they occur in synchronism with the cyclically moving object under observation. These light flashes are used to illuminate the mechanism that is being studied. To obtain maximum illumination from the light flashes, the light is projected in a beam by a suitable reflector. With this type of instrument, several observers can study simultaneously a moving mechanism.

Originally, only a glow lamp fed by condenser discharges was susceptible to such flashing service. Because of its low electrical inertia, the glow lamp made it possible for the first time to emit light flashes of a duration in the order of 0·00001 second. The highest illumination is obtained from lamps filled with a mixture of helium and neon, but owing to the spectral nature of such gases, the emitted light is reddish and is not very useful for observation and photographing.

In recent years, glow-lamp type stroboscopes have been considerably improved. The gas-filled discharge lamps of modern instruments contain separate ignition electrodes and cathodes. They are the most widely used, reliable and economical stroboscopes. Although the light intensity of a good glow-lamp stroboscope is not higher than that of a 40-watt bulb, it is sufficient for a great number of movement studies, because the whole amount of emitted light is focused towards the point of inspection and made useful for clear observation. However, the general or ambient illumination, whether artificial or natural, upon which the stroboscopic illumination is super-imposed, may mar a clear image.

With a distance of one metre between the stroboscopic lamp and the object under observation, an ambient light intensity above 100 lux will impair the stroboscopic image, which will be seen as a faint shadow of a uniformly blurred background image of the moving object. The background image is, of course, caused by the general illumination. To overcome this, an increase in stroboscopic light intensity was effected by the introduction of the xenon flash lamp. This type of lamp has found many applications.

### Xenon lamps

A xenon flash lamp is essentially a gaseous chamber comprising a helical glass tube of two to five coils arranged substantially about the focal point of a reflecting mirror. Electric discharges of about 30 to 50 cm length are effected in the chamber. The gaseous filling of the chamber consists generally of a mixture of crypton and xenon or of pure xenon at a pressure of approximately 0·1 atmosphere.

Owing to the great effective length of the discharge chamber, it is possible to subject a xenon lamp to a relatively high discharge loading that gives flashes of much higher light intensities

than those of normal glow lamps. In addition, because of the spectral nature of crypton and xenon, the light has a yellowish-white or bluish-white colour. The average duration of the single flash is in the order of 1/30,000 second. In combination with its reflector, the lamp gives a light intensity about equal to that of a brightly-burning 100-watt bulb. This allows good observation in plain daylight of a periodically moving object at a distance of half a metre from the stroboscope.

With this type of instrument exact speed measurements and slip determinations can be made. There are instruments that give an accuracy of better than plus or minus one per cent in determining frequencies. Since 1945 the use of xenon lamp stroboscopes has continually increased. They have been used successfully for studying valve springs in respect to surging and super-oscillation, for slip determinations, for observing and adjusting contact springs against bouncing effects, and for many other applications.

However, the xenon lamp stroboscope has its limitations, especially in respect of the intensity and the duration of the flashes. The light intensity is too low to effect a photographic exposure by a single flash; the duration of the flash is too long to give a clear image if the object is moving at high speed. If the velocity of the object is 30 metres per second, the motion blur amounts to as much as one millimetre. More recently, instruments have been developed with higher intensity and shorter flash duration. They are known as high-intensity rapid flash stroboscopes or as "super stroboscopes."

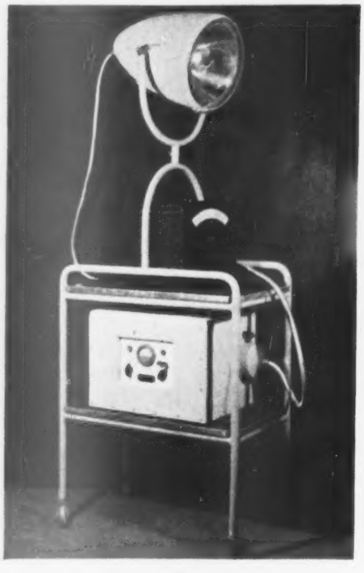
### Super stroboscopes

Two basic systems have been developed for "super stroboscopes." In one, a relatively large condenser is discharged through a capillary lamp filled with argon by means of an ignitron. The argon capillary produces a very intense light flash of from three to five micro-seconds duration at each discharge. The light is optically converged to a slightly spreading beam to give flash intensities that permit good photographic exposures by single flashes on normal film material at a lens aperture of f.11. Unfortunately, on account of inherent physical properties, the inertia of the ignitron causes time lag, and thereby creates difficulties, because even with the most accurately timed control pulses, an exact recurrence of light flash initiation cannot be assured.

Variations in flash timing create a fuzzy and fluctuating image. To overcome this disadvantage, systems with an auxiliary discharge path in the stroboscopic lamp have been introduced. But even this mode of operation is not free from drawbacks, because a certain amount of light is carried over between the flashes. It is sufficient to impair the quality of a photograph taken at a single flash if the camera shutter is opened in advance.

A great advantage of this system is that the complete flash lamp with reflector and capillary tube can readily be exchanged or replaced. A disadvantage is that the lamp has a relatively short life, which is further reduced by diminishing light emission as a result of deposits on the inner walls of the sealed envelope that arise from the unavoidable heavy currents of the flash discharges.

The second type of "super stroboscope" uses a demountable spark-discharge vessel as the light source. A pair of replaceable rugged tungsten electrodes are mounted in the vessel. The walls of the vessel, formed by a glass cylinder, can easily be cleaned from inside and outside. Provision is made for maintaining the gas pressure at a suitable value within the spark-discharge vessel. Whenever necessary, the gas can be replenished from a supply flask filled with commercial argon.



High-intensity, rapid-flash stroboscope

A bank of condensers mounted directly in the back of the discharge vessel produces the flashes by discharging them through the gap between the electrodes. The flashes, which result from spark-overs, are initiated by ignition sparks that originate from voltage pulses generated, together with the high voltage for charging the condenser bank, in a power pack fed from a general current supply source. An oscillator, serving as a control unit, governs the ignition pulses so that the flashes occur in accurate synchronism with the frequency set on the control unit dial. The discharge mechanism allows for an accuracy of flash initiation that strays less than $10^{-7}$ second from the theoretical. A surprisingly short flash duration is obtained. It is photographically effected for less than one micro-second.

These exactly timed flashes of extremely short duration give sharply defined and steady images. The spark-overs give a light of bluish-white colour of sufficient intensity in the direction of the beam for getting an exposure by a single flash on normal film material at a lens stop of f.11. A slight disadvantage of this system is that the demountable discharge vessel needs some attention during service. This servicing is, however, quickly carried out.

Recently, this system has been improved by connecting directly to the control unit a small glow-type stroboscopic lamp that flashes in synchronism with the ignition pulses. Briefly, the auxiliary glow lamp can be used to illuminate, stroboscopically and dimly, the object to allow adjustment until the desired phase of movement is obtained. This method eliminates the necessity for trial and error and provides for the first time means for recording the desired phase of motion on the first picture and of following through a motion study in systematic order.

### Flash durations

A "super stroboscope" with flash durations between one millionth and one-ten-millionth of a second opens completely new vistas for sharp photographic records of fantastically fast motions. Stroboscopic analysis becomes possible for a whole new realm of problems in dynamics. Some typical applications may serve to illustrate the important function of a high-intensity rapid-flash stroboscope in manufacturing and research.

In tyre production, tests on new designs are carried out at high rotational speeds. The revolving tyre can be made to appear stationary by stroboscopic means. Then, by letting the thread strike a bar it is possible to observe continuously all the deflection round the calks of the tread. Developing defects can be seen, and the life expectancy of the tyre can be predicted after a relatively short test run. As such runs are conducted at inconstant speeds, it is necessary to trigger the lamp flashes by a precision contacting device. This device effects periodic flashes that recur once for every revolution of the tyre with an accuracy of a few angular minutes. The flashes can be adjusted for viewing the tyre in any one of its angular positions during rotation.

In dealing with injection pumps, it is important to determine the size and form of the atomized droplets in the spray from the nozzle. Velocities up to 200 metres per second are encountered in the issuing spray, and even a light flash of one micro-second duration causes a motion blur of 0·2 mm. As the size of the largest droplet is of the same order as the motion blur, the flash duration is too long. Nevertheless, it is possible to distinguish whether a droplet is stretched lengthwise or has the original streamlined shape.

Efforts are now being made to reduce the photographically effective flash duration in order to reduce the motion blur to an even greater degree of high-velocity minute particles. The difficulty of the problem is shown by the fact that even now stroboscopes are available to give a photographically effective flash duration in the order of $0{\cdot}4 \times 10^{-6}$ seconds.

In many applications a high-intensity rapid-flash stroboscope can be used to advantage in conjunction with an accurate contacting device. This combination makes it possible to take a film recording once each revolution at a certain phase of the periodic motion; for example, of a piston rod at the point of reversal. The recordings of such motions are superimposed on the film and are taken at various machine speeds. The contours of the recording give an indication of the clearance condition of a specified machine element. If further similar recordings are taken after the machine has been in service for some time, conclusions concerning changes in clearance and concerning wear may be drawn from comparison of the two recordings.

#### Stroboscopic photography

For ultra-rapid continuous film photography, the high-intensity rapid-flash stroboscope supplements high-frequency ultra-rapid motion picture apparatus. All that is needed as a camera is a rotating drum for the film with a lens in front of it. If 300 light flashes are triggered per second, 300 exposures or frames are obtained per second. Owing to the micro-second duration of each flash, each frame is exposed for only one micro-second. Therefore, there is no motion blur even when ultra-high speed phenomena is being recorded.

The extremely short exposure time made possible by stroboscopic illumination has the advantage that even at a low rate of frames per second, sharper images of high-speed motions can be obtained than with normal ultra-rapid motion picture apparatus, which operates with longer exposure times that cause an undesirable motion blur. A modification of the high-intensity rapid-flash stroboscope, designed specially for ultra-rapid motion picture photography, employs flowing noble gases and permits exposures at the rate of 3,000 frames per second with correspondingly short exposure times.

To obtain fully exposed shots with ultra-rapid motion picture apparatus, a bright light source is of primary importance. In many applications such intensive light sources have caused trouble on account of undesirable and harmful heating effects created on the object by the light source. There is no such trouble when a high-intensity rapid-flash stroboscope is used as the light source for such work. The bluish light of the spark-overs gives abundant illumination of high photographic value without excessive heating of the object. Also, owing to the micro-second duration of the flashes, it gives pictures of great sharpness. Even a camera without film-feed mechanism and shutter, but with only a rotatable film drum will suffice.

Measurement of the inherent vibrations of machine elements can be simplified by the use of a high-intensity stroboscope, in combination with microscopic observation. For example, the inherent vibratory frequency of a turbine blade can be measured in the following manner. The turbine blade is excited in its own frequency, and the frequency is picked up by an electro-magnetic transducer to be amplified. By means of a frequency divider every hundredth half-wave of the vibration is picked out and transferred to the stroboscopic control circuit. For an inherent vibratory frequency of 5,000 cycles per second, there will be 50 light flashes per second. With a flash duration of $1 \times 10^{-6}$ seconds, microscopic observation will present sharp images with a motion blur of less than 2 deg. This 5,000 cycle vibration can be observed throughout its whole transition by means of a phase shifter. This method opens new ways for investigating ultrasonic vibrations with magnetostrictive excitation of optically visible harmonics, or for photographic recording of cavitation proceedings of a single oscillation. New aspects of cavitation research are presented.

The behaviour of gears under load can also be studied stroboscopically. Inaccuracies in tooth form, which may cause irregularities, power losses and excessive noise, can readily be traced. For these investigations, the light flashes are synchronized with a specific gear tooth so that the whole rolling-off behaviour of the tooth flanks may be studied. Stroboscopic examination in combination with microscopic observation also makes it possible to detect torsional vibration which, if present, affects the true running of the gears by breaking the contact between the flanks in certain spots.

#### Torque measurement

With a high-intensity stroboscope it is a relatively simple matter to measure the torque in shafts or clutches during actual operation. Two measuring discs with suitable graduations are mounted in a certain axial spacing on the shaft. By an arrangement of prisms, somewhat similar to that employed in geodesy, the two measuring discs can be brought into coincidence in the view field of a telescope. Before the actual operating test is carried out, a static torque calibration test must be made to determine the torque corresponding to a certain deflection of the shaft. The machine is then operated at various speeds and the shafts and measuring discs are stroboscopically illuminated. This allows the relative displacements of the graduations to be observed through the telescope, and from the obtained values the actual deflection of the shaft can be determined.

Clutch slip can also be measured in a similar manner by observing, either visually or photographically, one spot on the clutch periphery. Not only the slip of clutch halves, but belt slip also can be exactly measured. This procedure provides scientific methods for determining by optical means friction losses in drives and clutches. Valuable information can be gained concerning undesirable inherent vibrations, which are particularly likely to occur in steel clutch discs. The cause of failures in such transmission elements, which previously could not be traced, can now be readily discerned by high-intensity stroboscopic examination, since excessive wear and premature failure, especially in clutch discs, are the results of vibration.

#### Slow motion analysis

In automatic assembly machines the periodic repetition of any phase of movement is often very slow, in the order of two to three cycles per second. At a flashing rate of two to five per second, it is not possible to obtain visually the illusion of a steady image. To make an analysis of such slow movements and their kinetic progression, photographic means have to be applied in conjunction with a stroboscope. For satisfactory results it is necessary to take a series of photographic shots by the illumination of a stroboscope with flashes triggered by a contactor actuated from a suitable shaft on the machine. To record the phase progression of the movement, the contacting instant of the contactor must be continually changed from flash to flash by a suitable amount, say one angular degree. A suitable contactor for such work has been specially designed. It has a conical friction drive with a ratio range from 1 : 0·9 to 1 : 1·01 that automatically moves each successive flash in its time relation to the previous flash. The whole cyclic movement is therefore, recorded in 100 phases if 100 flashes are triggered in steadily accelerated or decelerated time relation. For this type of recording, an automatic film feed camera with electrical operation should preferably be used.

In the study of air-flow physics, the high-intensity stroboscope can be employed in conjunction with the well-known light sectioning method, whereby aluminium tinsel is admixed in the air flow. Because of the extremely short duration of the light flashes, the photographic recordings are very exact and reveal all details concerning magnitude and direction of air flow.

A high-intensity stroboscope can also be used for detecting inhomogeneities in the rotor structures in large electric machines. In studying an electric motor, the light flashes of the stroboscope should be synchronized in relation to the number of poles of the machine. Images revealed during such observations have disclosed surprising facts concerning the reason for vibration in such machines, which have been balanced statically and dynamically, but still tend to be noisy in running. It has been found that the cause of such faults is that some poles project

beyond their theoretical limits and give rise to undesirable inherent vibrations.

Brush vibration in large direct-current generators and motors operating at high peripheral commutator speeds, can be exactly studied by means of the high-intensity stroboscope. It is generally caused by insufficient damping of the brush holders. It is often found that unexpectedly heavy firing of brushes, despite painstaking mechanical fitting, is caused by incidental resonance between the inherent frequency of the brush holder and the frequency of the commutator segments. Once detected, this can readily be remedied by shifting the resonance or by damping by applying additional rubber cushioning. In passing, it may be mentioned that a high-intensity stroboscope may be used advantageously for producing photographic records of the balancing phases in dynamic balancing machines.

If a high-intensity rapid-flash stroboscope is to be applicable to the applications enumerated, it must be provided with some means of selective flash triggering. Flash release can be controlled by hand or mechanically either by contact making or contact breaking, or by electric pulses of an oscillator whose frequency can be varied over a wide range. Above all, no matter what kind of triggering is applied, it is mandatory that the intensive flashes of the lamp are initiated in correlation with the governing control means.

Stroboscopic work in conjunction with photography is considerably simplified by applying an auxiliary stroboscopic glow lamp. If a setting is made for a certain phase of movement by the light of the glow lamp, there is assurance that the photographic shots will be exact recordings of the desired phase movement.

A contactor must be dependable and of very high precision. It must react in absolute dependence upon the angular position of a rotating actuating shaft. Frequently it is necessary, especially when acceleration is being measured, that the contactor takes the function of divisional indexing to ensure that the flashes occur at equally spaced angular intervals in the complete revolution of the shaft. Under very unfavourable optical conditions, it may be necessary to connect an additional external condenser in the circuit of the stroboscopic light source in order to strike even brighter single flashes.

Although the high-intensity, rapid flash stroboscope is generally applied to the study of systematic problems, its operation requires intensive training and skilled handling. The "super stroboscope" is not an instrument for regular production control. It is an instrument for scientific investigations and can be used for analyses of dynamic phenomena that hitherto have been beyond the limits of investigation. The high - intensity stroboscopic equipment referred to in these notes is manufactured by Physical and Technical Laboratories, Dr. F. Fwrengel, Hamburg, Western Germany, and is obtainable through Aga Standard Ltd., 16-17, Devonshire Square, London, E.C.2.

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# TURBO-CHARGERS FOR DIESEL ENGINES

THE difficulties in the application of turbo-chargers to motor vehicle diesel engines are discussed in *Diesel Power*, August, 1952, issue.

In most power applications, engines operate at relatively constant speed and, in meeting power demands, the only other requirement is sufficient torque. Load changes are generally fairly slow, the torque curve being relatively flat in the normal operating range. Turbo-chargers vary their output automatically with variations in engine load and speed, and where the only essential variable is load, the turbo-charger adjusts its output to relatively slow load variations. With motor-vehicle diesel engines, the operating range is much wider and conditions may vary from low speed-high torque to high speed-low torque. Since the turbo-charger responds to both engine load and speed, the difficulty is to produce a design to cope with all the varying conditions. For acceleration purposes, the engine must be capable of increasing its power more rapidly than the load requirement increases with speed, smoke-free operation being essential under all conditions.

Small high-speed engines have been successfully turbo-charged to give an increase of 25-50 per cent over their normally aspirated ratings. However, increase in power output and reduced fuel consumption are not enough to make a given engine acceptable for motor-vehicle use. The solution to the acceleration problem depends on the basic design of the turbo-charger rotor and to some extent on the way in which the turbo-charger is combined with the engine.

The Cummins Engine Company's turbo-charged diesel-engined racing car is fitted with a special Elliott turbo-charger. The design combines minimum rotor inertia with maximum performance from both turbine and blower, satisfactory automotive torque characteristics, and materials resistant to high engine exhaust temperature. Casings for the blower air flow and turbine exhaust gas flow, located within a few inches of each other, withstand temperature differences exceeding 1,000 deg F. The turbine used is of the radial in-flow type. Pressure-lubricated sleeve bearings permit operation above 36,000 r.p.m. The record-breaking runs of the Cummins racing car prove that a turbo-charged engine can give good flexibility, acceleration, and satisfactory performance. (*M.I.R.A. Abstract No.* 6044.)

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# DETECTING LEAKS IN HOLLOW COMPONENTS

THIS article by L. E. Sterns in *Automotive Industries*, September 1, 1952, gives what is claimed to be the first published description of a new technique of testing hollow components for leaks by means of vacuum. The equipment required includes a pump, capable of producing a vacuum of 28 to 29 in, with sufficient capacity to evacuate rapidly the space to be tested, a fixture to which the part is applied in such a way as to make a perfect seal by a pad of neoprene, soft rubber or the like, a vacuum valve, connecting line and a sensitive vacuum gauge.

The method of operation is to apply the part to be tested to the fixture, draw out the air, close the valve and watch the vacuum gauge. Any falling-off in the gauge reading indicates a leakage of air into the system, quite small leaks being discernible.

Automotive parts which may be tested in this way include cylinder heads, cylinder blocks, gear-boxes, rear axle housings, ground-in valves, oil seals, sumps and water-pump assemblies, the method being ideal for detecting porosity in castings and the like. Light gauge parts such as petrol tanks can also be tested, but at reduced vacuum on account of the risk of collapse.

The popularity of vacuum testing is attributed to the fact that it is fundamentally simple, clean, very productive, safe, and low in cost per part tested. (*M.I.R.A. Abstract No.* 6067.)

# CRANKSHAFT PRODUCTION

## *Equipment and Methods Employed by Ambrose Shardlow & Co. Ltd.*

CRANKSHAFT production as practised by Ambrose Shardlow and Co., Ltd., Sheffield, raises problems that differ considerably from those dealt with in automobile factories. There must be provision for dealing with a wide range of designs required in widely varying quantities. Some crankshafts are needed in quantities large enough to warrant heavy outlay on tooling; others in quantities so small that tool costs must be kept as low as possible if the unit cost per shaft is to be reasonable. At the same time, the high standard of quality that this organization specifies for its products must always be maintained.

Adequate machine utilization, a major factor in manufacturing economy, is inevitably difficult in production that involves short runs and frequent changes of set-up. It is, of course, possible to obtain it by employing a large production control department but without any guarantee that the economies obtained through good machine loading may not be offset by the extra non-productive labour costs. Ambrose Shardlow and Co., Ltd., has, as a result of many years' experience, built up a system that ensures economical use of plant though there is only a relatively small production control department.

Throughout the organization the practice is to keep the number of non-producers to the efficient minimum. For example, an inspection staff that is small in relation to the volume of work produced serves to maintain a very high standard of product quality. In general, this has been effected by employing special equipment with accuracy dependent upon the machine rather than upon the care and skill of the operator. This is particularly true of the machine shop, where the equipment is such that patrol or stage inspection is practically unnecessary,

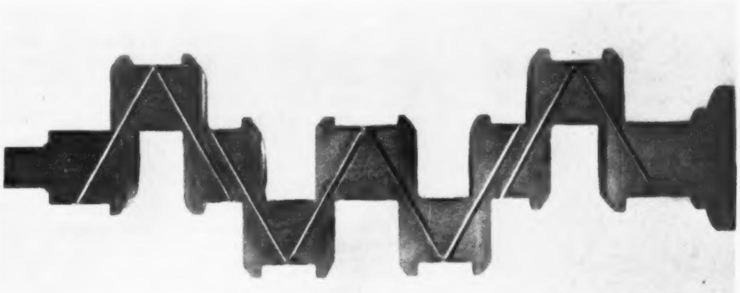

Fig. 1. Sectioned crankshaft, etched to show grain flow

and the inspectorate is able to concentrate almost solely upon the quality of the finished product. In the forge, manual skill remains much more important, although even in that department, every available means is used to eliminate, or at least minimize, the possibilities of human errors.

### The forge

The forge organization comprises the following sections:—

(1) Die designing and planning offices.
(2) Die shop.
(3) The forge.
(4) Heat - treatment, fettling and inspection sections.
(5) Metallurgical department, which also serves the machine shop.

Die design has an important influence on all the forging operations, since it affects both the cost of production and the quality of the product. There is a difference in die procedure as between dies for crankshafts required in large or relatively large quantities and those for shafts required in small quantities. The practice for large quantities will be considered first.

In the forging of Shardlow crankshafts, the special feature is the amount of work at the "use" or pre-forming stage. The "use" is produced in a "blocking" die, which may have either two or three impressions. Generally, there are three, one for rolling, one for bending, and between them one for "blocking." At the first or rolling impression the billet is manually rotated and circumferential ribs are produced. The bending impression imparts a folding-type deformation of the billet in a vertical direction parallel to the downstroke blow of the hammer.

At the blocking impression the billet is laid on its side so that the hammer blows force the material to flow horizontally and at right angles to the direction of the stroke. The effect of blocking is to spread and move material

Fig. 2. Part finished blocking die and, at the rear, single impression finishing die

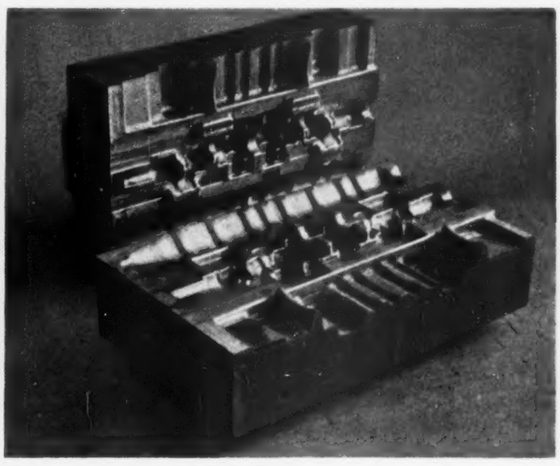

Fig. 3. Three-impression die block for operation on a 20,000 lb Chambersburg hammer

Fig. 4. Producing a hand-made use

increases the life of the finishing dies and reduces hammer wear. But the blocking operation necessitates special precautions. Two of the three impressions are offset in relation to the centre line of the hammer, and the hammer must be reinforced to withstand heavy side stresses.

In every case particular attention is paid to the balance of the die in the hammer by attaching it to the tup so that there is as nearly as possible an equal amount of work on each side. The use is also balanced to give equal pressure and flow in the finishing die, a uniform thickness of flash and uniform cooling. Through the care taken in design and use of dies, the blocking dies have

to where it is required. It gives a grain flow that is markedly radiused and streamlined without sharp angularity. This is shown in Fig. 1. In practice the hammer operator rapidly alternates the "use" between the bending and blocking impressions. A blocking die in course of preparation is shown in Fig. 2 in front of a single impression finishing die.

The blocked use is symmetrical about its longitudinal axis. Therefore, when it is transferred to the finishing hammer, after re-heating, it does not tend to cause the tup to tilt. This eliminates the risk of offset forgings,

a life equivalent to the production of 100,000 items or more. Finishing dies usually produce from 4,000 to 6,000 forgings. The latest development in die design is the use of a three impression block on a Chambersburg 20,000 lb. hammer. This combines the blocking use procedure and the finishing impression as shown in Fig. 3.

A very different procedure must be used for crankshafts required only in small quantities. The expense of blocking dies is not warranted, and the uses are partly hand forged in that hand-operated tools are used under a power hammer, as shown in Fig. 4. At

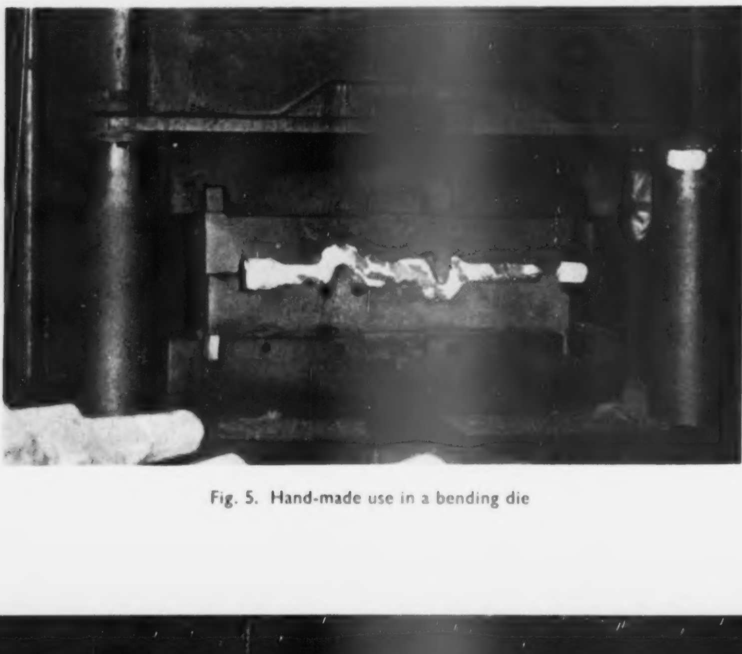

Fig. 5. Hand-made use in a bending die

the same heat, these hand-made uses go through a bending operation, see Fig. 5, but there is no "blocking." Normal finishing dies are used. All clipping tools are designed to fine limits to give a close finish. Stellite is deposited along the cutting edges and the output per tool is in the order of 15,000 units.

Dies are very expensive tools, and naturally the greatest care is taken to ensure correct design. The die drawing goes through a number of stages. The rolled form is superimposed on the drawing of the finished crankshaft. Bending and blocking lines are then drafted between the rolled form and the final form, according to calculations of the amount of metal to be moved at each heat and by each die impression. Often, wooden models of the crankshaft and the use at various stages are made up, and the die impressions are set out in plasticine to give indication of the flow properties to be expected at the different heats.

### The die shop

The die shop is inset into the machine shop. It has been located away from the forge to ensure freedom from the vibrations that are inseparable from the operation of heavy hammers. It is fully equipped for the production of every type of die that will be required. The machine tools include a large and a small Stirk "Hiloplane," two Cincinnati "Hydrotels" and four Keller die-sinkers. One of the Keller machine is shown in Fig. 6.

Die blocks vary in weight from 15 cwt to 4 tons and may measure up to approximately 63 in × 33 in × 20 in. Those for making uses, including blocking dies, are usually of carbon steel, as are short-run dies. Finishing dies are made from proprietary steels of the nickel-chromium-molybdenum type. These blocks are received and machined in the 70/77 tons tensile condition. High speed steel tools are used. An example of the care for economy that is typical of this organization is that planer tools when their machine life is finished are forged into chisels to be used for hand die sinking. Standard Herbert tools are used on the Keller machines.

Every die is, of course hand finished by a highly skilled craftsman. Shrinkage allowance is generally in the order of $\frac{1}{8}$ in per foot, but for the higher rates of output it may be increased to $\frac{3}{16}$ in per foot to compensate for the higher temperature of the die. Taper of 5 deg or more facilitates ejection of the work from the die. It is seldom that a Shardlow die cracks or breaks in service, evidence of the skill used both in designing and making the die.

### The forge

For very good reasons, the forge is completely separate from the remainder of the Shardlow works. It is housed in a group of buildings that include three hammer bays, maintenance, heat treatment and fettling

departments, and power and compressor houses. It is not necessary to describe the forge layout and equipment in detail.

In general, it may be said that the plant in two of the bays was laid out for relatively long run work when it was installed, but it does not follow the most recent practice to which we refer later. When necessary, short run work has to be carried out on these same hammers. The largest hammer in use at present is in the second bay. It is a 20,000 lb air-operated Chambersburg hammer. On this hammer, the crankshaft can be produced direct from the billet in one die that combines the rolling, bending and finishing impressions. In this case, the blocking operation is not necessary because of the extra work that is put in on the other two use making operations.

The third bay is representative of most advanced practice for the production of crankshafts. It has the following plant laid out for line production.

(1) A Gibbons gas-fired pusher-type furnace.
(2) A 7 cwt Massey hammer for tagging.
(3) A 7-ton Fielding and Platt hammer for producting the use.
(4) A 5-ton Fielding and Platt hammer for finish forging.
(5) A 1500-ton Wilkins and Mitchell clipping press.
(6) A 3-ton Chambersburg hammer for coining and straightening.

With fast movement between them, it is possible to carry out all the operations at one heat. A system of overhead runways has been installed to allow the work to be taken round the "U"-shaped circuit at the necessary speed. This form of layout will be extended as circumstances warrant to other parts of the forge.

There is a separate shop for hot twisting crankshafts with the pins and webs not stamped in the correct position. In cases where the six throws are at 0, 120 and 240 deg, two pins are held at 0 deg while two are pulled to 120 deg. The shaft is then repositioned in the jig and the other two are pulled to 240 deg.

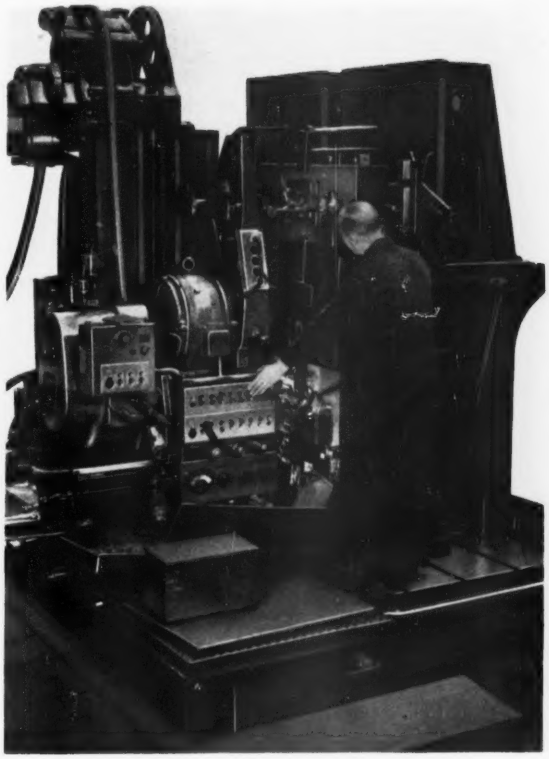

Fig. 6. Machining a die block on a Keller die-sinker

### Heat treatment and fettling

Three factors determine the heat treatment procedure for a given shaft; the composition of the steel, the mechanical properties required, and the previous history of the forging. Carbon steel forgings are ground (fettled on the flash line), normalized, hardened, tempered, brinelled, and most usually second tempered. The normalizing regularizes the structure at different parts which may have undergone unequal heating and chilling; it may also be used to produce a uniform structure in forgings that have been heated more than once in production. Normalizing is not invariably carried out on carbon steel forgings, some of

Fig. 7. A battery of Gibbons furnaces for heat-treating forgings

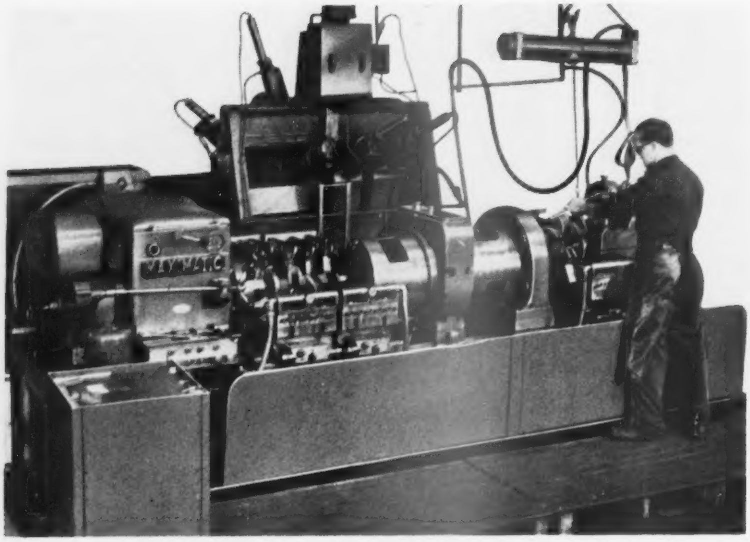

Fig. 8. Drummond Maximatic centre drive lathe for turning all main journals, web faces and bevels and timing wheel and flange diameters

which go direct to hardening and tempering. Every shaft is brinelled for hardness after the tempering operation. To ensure correct readings, a small portion of the forging is ground to remove scale and the outer skin. Should it be necessary to hot straighten a shaft, this is carried out immediately after the tempering and is followed by a second tempering at a lower temperature than the first. The hardness test follows.

Alloy steel crankshaft forgings follow a different procedure. They are hardened, tempered and brinelled before they are fettled. At least one subsequent tempering is carried out to relieve fettling stresses and, if the operation has been necessary, hot straightening stresses. Many alloy steel crankshafts are subject to further tempering. Tempering operations are carried out at successively lower temperatures. High alloy shafts, and particularly those for nitriding, are heat-treated after rough machining to relieve any machining stresses and to ensure permanence of dimensions at finish machining. The actual heat treatment sequence for any shaft is greatly dependent upon the forging practice that is employed. For example, a shaft that is produced at a single heat in a three impression die does not raise the same heat treatment problems as for one that is re-heated once or more during the forging sequence.

The heat treatment, fettling and inspection shop is a large and very clean and orderly department for work of this type. Heat treatment is carried out in either Incandescent Heat or Gibbons furnaces. The oil and water quench tanks are, of course, placed conveniently in relation to the furnaces. Fetting is carried out on swing grinders, pedestal grinders and portable equipment in a position that calls for little movement of work from the heat treatment section.

Of the furnace equipment, attention may be drawn particularly to Gibbons furnaces, see Fig. 7, that have recently been installed. Each furnace has a hollow roof with a perforated refractory ceiling over the hearth. Pre-heated regenerated air at 400 deg C. is delivered into the roof cavity, and gas is supplied through a number of side jets below the ceiling. The air is drawn into the gas area and combustion takes place below the ceiling. A controlled furnace atmosphere ensures complete freedom from oxidization. Easy and correct operation is obtained through pressure and mixture controls that are incorporated in the furnace design.

### A typical forging sequence

A typical forging sequence for a six-throw diesel engine crankshaft is:—

(1) Preheat to 1250 deg C.

(2) Die-forge use on blocking hammer.

(a) Tag on end of die, then pass through die to the stamper.

(b) Rotate in roll impression on one side of die.

(c) Form in bending impression on other side of die.

(d) Strike in central, blocking impression.

(3) A number of blows with the use alternated between 2c and 2d.

(4) Reheat.

(5) Finish forge in finishing dies.

(6) Clip hot.

(7) Return to finishing hammer for final tap.

(8) Inspect.

(9) Heat treat and fettle.

(10) Final inspection.

### Technical department

Well-equipped chemical and physical laboratories are important sections of the technical department. In addition to tensile testing and Izod impact machines and other equipment normally used in a physical laboratory, there is also a full size fatigue tester for alternating bend tests. This department is responsible for checking the quality of incoming material and for comprehensive daily routine tests on specimens taken from representative cranks.

Metallurgical technical control is also exercised by the technical department, as is experimental work to ensure that the forging procedure for a new design of crankshaft will be such as to produce good grain flow.

From the receipt of the incoming materials to the transfer of the forged shaft to the machine shop, every possible care is taken to maintain the specified high standard of quality. For this reason, twice a day the production and technical management assemble at the forge headquarters to carry out a detailed examination of a crankshaft or use chosen at random from the production from each hammer. Every conceivable point of interest from metallurgical and dimensional factors to die life is given full consideration.

### The machining division

The machining division includes:—

(1) Drawing office.

(2) Planning department.

(3) Machine shop.

(4) Tocco and nitriding departments.

(5) Balancing, lapping and finishing sections.

(6) Jig and tool shop.

(7) Instrument and gauge section.

The main machine shop building is approximately 430 ft long and 390 ft wide. It has 13 longitudinal bays, each 35 ft. wide. These bays are bisected by a transverse gangway, so that the shop is divided into 26 areas. Each bay has two rows of machines, separated by a wide gangway. Most of the bays

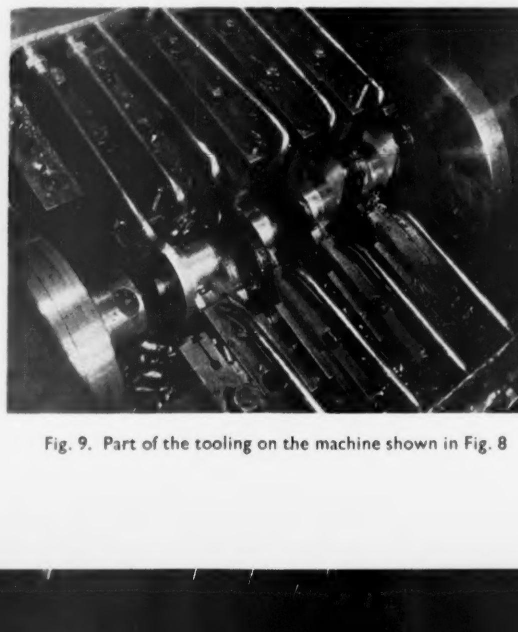

Fig. 9. Part of the tooling on the machine shown in Fig. 8

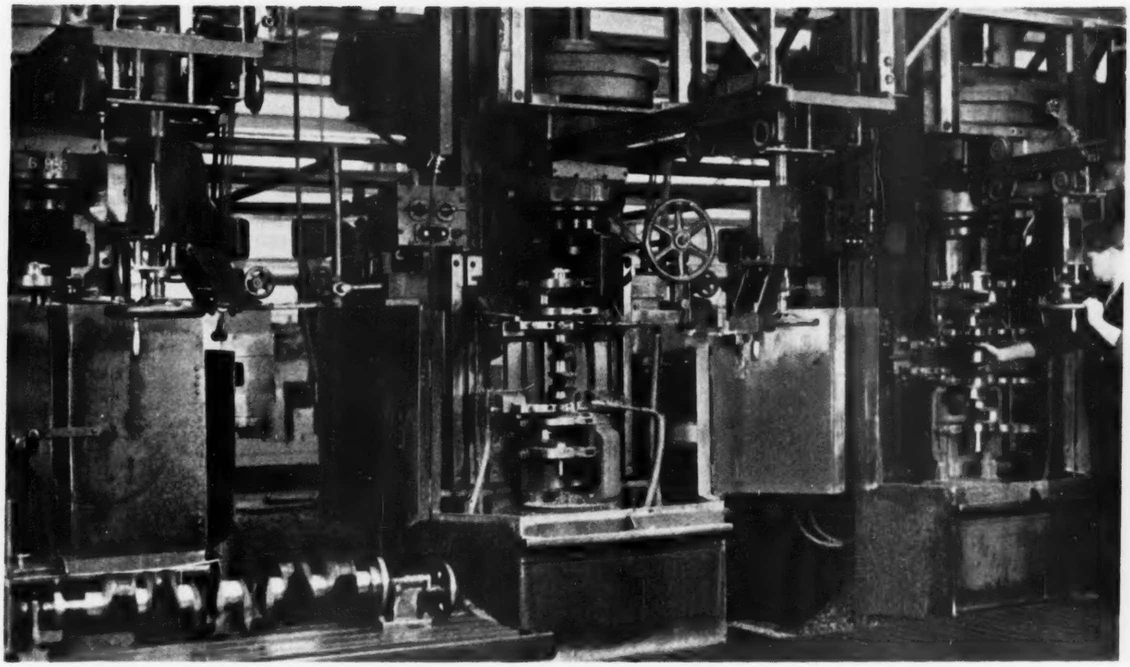

Fig. 10. Special Shardlow vertical crankpin turning machines

are served by overhead cranes running the whole length of the shop.

Several methods of materials handling are employed. At the machines themselves, handling is generally effected by means of mobile air hoists or block and tackle mounted on runways. For movement between sections there are special purpose hand trucks and power platform trucks, but most of this type of work transfer is effected by slinging a batch of crankshafts from the overhead crane. The shape of a crankshaft is such that when a number are stacked they will interlock sufficiently to give a stable stack that does not require any extraneous support. Shafts are therefore stacked to a height of five or six feet convenient to the machine in which they are to be worked. Bins, stillages or racks are not necessary, but the floor is tidy, and work movement is effected by only a small labour force.

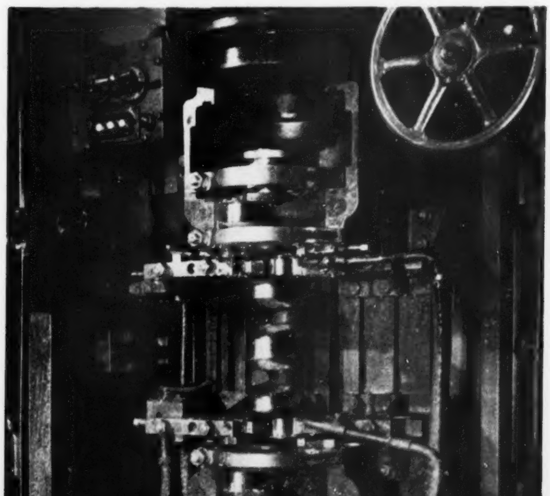

Fig. 11. A crankshaft in one of the machines shown in Fig. 10

Because of the wide variations in the quantities required of different designs, the machine shop is in part laid out for product grouping and in part for process grouping. That is, there is product grouping for long run production and process grouping for shafts that are required in only small quantities. For each type of production the most economic method is adopted. On long runs, product grouping gives very good machine utilization and reduces transport to a minimum. For short run work, process grouping necessarily entails more work transfer, but to offset this, it does allow the best possible machine utilization.

Many special-purpose machines are used for crankshafts required in large quantities; some machines, in fact, are built for only one type of crankshaft. As far as possible British machine tools are used, but there are also many American machines that have been specially designed for high quantity production. Included in these latter are Wickes and Le Blond crankshaft lathes and Landis grinders.

The efficiency of the plant and layout may be illustrated by describing the production sequence for a six-throw diesel engine crankshaft. To begin with there are three preparatory machining operations which give accurate locations for subsequent operations. At the first, each end of the shaft is centred in a double-end centring machine. The shaft is then transferred to a Norton grinder, on which the centre bearing is plunge ground, with suitable allowances left on the diameter and width for removal at subsequent operations. At the third, the flange is turned to a tolerance of 0·002 in in a Swift centre lathe. A high speed tool is used.

At this stage the shaft is transferred to a Drummond Maximatic centre drive lathe, tooled for turning all the main journals, web faces, web bevels, timing wheel diameter and flange

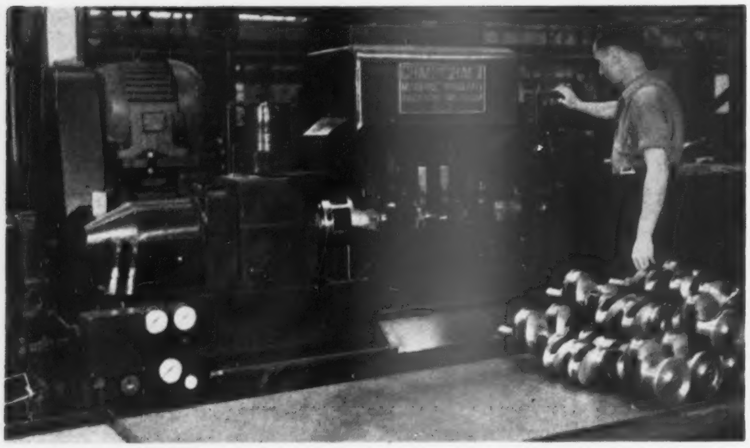

Fig. 12. Jackson lateral feed crankpin turning machine

diameter, see Figs. 8 and 9. This machine accommodates two crankshafts at once. The tooling is such that the timing wheel half of one shaft and the flange end half of the other shaft are machined simultaneously, so that the several elements are completed on one shaft at every machine cycle.

At the left hand end of the machine, the shaft is gripped on the centre bearing by nitride-hardened split bushes attached to the adaptor plate of a cast-iron barrel-type chuck which is deep enough to take half the length of the shaft. When the bushes are clamped together, the web adjacent to the centre bearing is brought up against a driving dog on the chuck face. A hardened live centre, adjustable longitudinally, supports the flange end.

The crankshaft in the other end of the machine is gripped in a chuck at the third bearing from the flange end to allow machining to be carried out on the centre bearing and the other three bearings towards the timing wheel end of the shaft, and also on the timing wheel diameter. An adjustable live centre supports the timing wheel end of the shaft. For ease of loading and unloading, the tailstock is air operated.

All the tool boxes, tool spacers and bushes are designed and made by the Company, and the tools are magazine-mounted in the toolroom. This ensures accurate spacing and facilitates setting-up. The tool boxes have milled dovetails to take the high speed steel dovetailed tangential tools. This allows the tools to be moved along the dovetail guides for adjustment of centre height after re-grinding. Cam-controlled, straight in-feed is used for both the front and rear tool boxes. The front box carries the tools for turning the bearings and bevel tools for machining a bevel on the webs; the rear box carries the tools for facing the webs. A speed of 30 r.p.m. and a feed of 0·010 in are employed. The tooling at one end is shown in Fig. 9.

The crankpins are then rough turned on special Shardlow vertical crankpin turning machines, illustrated in Figs. 10 and 11. Each machine turns two crankpin diameters and faces the appropriate webs in one cycle. On each machine there are two barrel type cast iron chucks, one attached to each face plate. Two adaptor plates are mounted in each chuck. A hardened split bush is carried in each plate, one half bush being carried in the adaptor plate body and the other in the locking clamp. These bushes are offset in relation to the vertical axis of the machine by an amount equal to the throw of the shaft. Therefore, since the bushes grip on the turned main bearings, these bearings revolve eccentrically while the pins revolve concentrically. Each machine also incorporates means for bringing the appropriate pins into the correct position for machining.

On the first of these three machines, the shaft is gripped on the outer bearings while Nos. 1 and 6 pins are turned. The second machine has extended chucks that allow the shaft to be supported on four bearings, to give rigidity while Nos. 2 and 5 pins are turned. On the final machine, the chucks are arranged to support the shaft on the four intermediate bearings while the centre pins are turned.

At each machine, four tool boxes are mounted on the slides, two at the rear and two at the front. The front tools plunge cut the pin diameter, while the rear tools, sometimes called finger tools, machine the web faces and collar diameters. High speed steel tangential tools are used. The tool slides are fed in by a hardened cam roller running in a cam track; as finished depth is reached, the continuous cam retracts the tool slides. At this stage, the shaft is checked for straightness, and if necessary, is straightened on a hydraulic press.

The main bearings and other elements rough machined on the Maximatic centre drive lathe are then finish machined in two operations on standard Maximatic lathes. The tooling is similar to that employed at the rough machining operations. After these finish turning operations, the shaft is again tested for straightness. It is then transferred to a Cincinnati milling machine on which location pads are milled. These pads are used to give accurate location in the machine used for finish turning the crankpins. These pads have no service

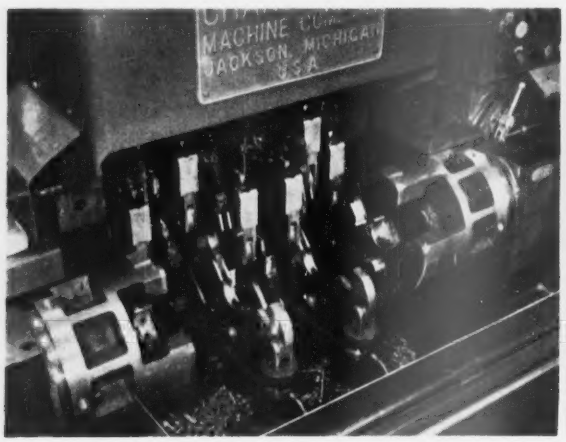


Fig. 13. The Jackson machine ready for loading

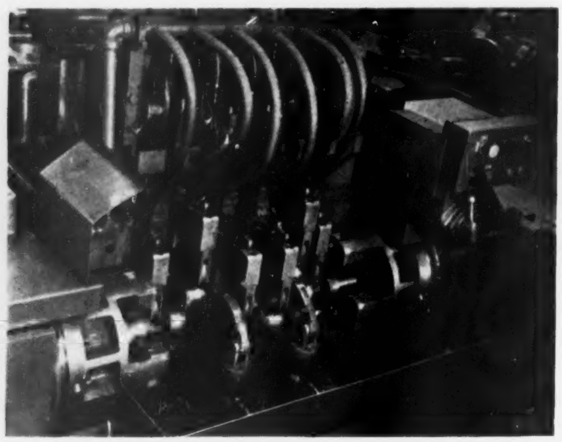

Fig. 14. The shaft in position for crankpin turning

significance, but they must be milled to very close limits of accuracy to give correct register at finish turning the crankpins.

There are, however, two further operations before the crankpins are finish machined. At the first, the timing wheel end of the shaft is finished by conventional methods on a No. 7 Ward capstan. For this operation the shaft is held on No. 7 bearing in a pot chuck while No. 1 bearing runs in a steady to give support near the points of tool application. From the capstan, the shaft is transferred to a Norton plain grinder on which the web faces of No. 6 crankpin are ground to width.

Crankpin turning is then carried out on a Jackson lateral feed crankpin turning machine, see Fig. 12. This machine is a recent development for the production of high quality work at high output rates. The outstanding feature of this machine is that all six pins are turned simultaneously. This may be called a copying machine inasmuch as the tool movements are controlled by a master template crankshaft mounted above the workpiece. The shaft to be machined is mounted in open-sided chucks at either end of the machine and the drive is taken from the milled pads. As the work drive and the drive for the master template are interconnected, the milled pads ensure that the work is in exactly correct relation to the template and that both work and template follow identical paths. To give support against the relatively heavy cuts that are employed, the work is also carried in three-point roller-contact steadies and clamps at Nos. 3 and 4 bearings, see Figs. 13 and 14. These clamps and the chucks are hydraulically locked.

Once the machine is loaded, the complete machining cycle is automatic. There is an initial fast approach to the cutting position with automatic change to feed rate. At a predetermined amount above the final dimension of the pins, the transverse movement is stopped and a longitudinal traverse introduced to machine the web faces by an amount that leaves a grinding allowance. At the end of the cutting cycle, the work is automatically returned to the central position, all reciprocating movements stop, the carriage withdraws and the chuck and steady clamps are unlocked ready for the work to be withdrawn and another shaft inserted. For a six-throw shaft the floor-to-floor time is in the order of 70 seconds.

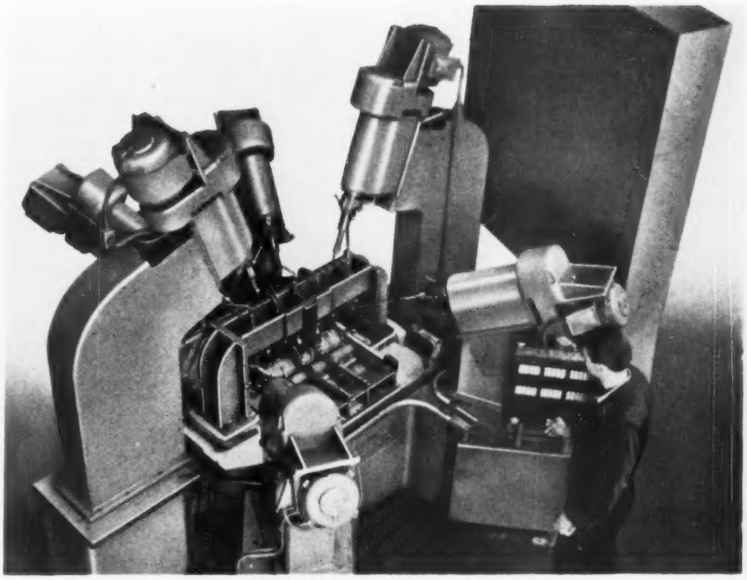

Fig. 15. Special Farmer automatic drilling machine

At this stage the angular oil holes from the main bearings to the crankpins are drilled. Two special automatic Farmer drilling machines, see Fig. 15, are used, each comprised of a central jig and six drill heads mounted at appropriate angles. The crankshaft is rolled into position under the fixture and the machine table is then raised to clamp the shaft into inverted vee blocks, see Fig. 16.

For the shaft under discussion, the drills are $\frac{3}{16}$ in diameter × 11 in long. They are pre-set to run at 800 r.p.m. with a feed of 0·0035 in per revolution. Hydraulic feed is employed, and by means of a time relay, each head is set to drill to a pre-determined depth and then withdraw. In addition, provision is made for automatic withdrawal of the drill if for any reason the torque exceeds a pre-set figure.

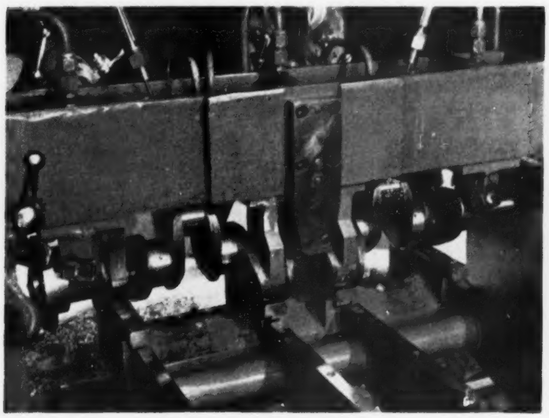

Fig. 16. The simple but effective work fixture on the machine shown in Fig. 15

Individual torque response is obtained through a Kirkman torque unit, which operates independently of, but concurrently with, the time relay; either one or the other may take effect, dependent upon which is first actuated. When the cycle is initiated, there is fast approach until a feed cam contacts a limit switch to bring in the feed rate. Drilling then proceeds and the cam bar advances with the drill, but the feed cam is halted at the limit switch. When the torque exceeds the pre-determined figure, the drill and cam bar retract. The feed cam also retracts with the cam bar, but in a position that corresponds with the depth already drilled so that on the next stroke the feed rate is not brought into action until immediately before the drill starts cutting. This sequence is repeated as often as necessary until the hole is finished and the time relay operates.

Tocco plant is used for hardening the crankpins. Part of this plant is shown in Fig. 17. There is a two-tier trolley track, the upper track for conveying the work along from heating station to heating station, while the lower is for the return of work-carrying trolleys from the end of the plant to the start point. The upper track is three feet from the ground and 50 ft long. It is covered by a sheet metal canopy that has apertures with sliding doors at the working stations. At each working station there are copper split-jaws connected to a high frequency circuit. The jaws are combined jigs, inductors and quenchers. They are machined to hold the appropriate crankpin or bearing on insulating pegs so that an even annular gap can be maintained between the metal surface of the jaws and the pin. The jaws are hollow and are perforated for the passage of the quenching water to the work.

At the start end of the plant, the crankshaft is placed on the work-carrying trolley, which is then raised by hydraulic lift to the upper track.

The trolley then passes along the plant and a different bearing surface is hardened at each station. Cycle time is from 15 to 20 seconds per station, comprising five to six seconds for heating and 10 to 15 seconds for quenching. Although this is a seven-station plant only six stations are operated for this particular crankshaft. The output is in the order of 20 crankshafts per hour. There is complete automatic control of all the factors that affect the quality of the work; consequently there is great uniformity of product. From the Tocco plant, the shaft is transferred to a slat conveyor tempering furnace.

**Finish machining**

Three lathe operations follow. At the first of these, the boss on the flange end is parted off and the end is re-centred. A Drummond Maximinor lathe is then used to face the back of the flange, rough and face the splasher diameter, chamfer the rear of the splasher collar, and groove and finish form. At the third, the oil splasher is screwed.

The number of grinding operations is governed by the variations in the widths of the bearings. On the crankshaft being discussed, the timing end, flange end and centre bearings differ from one another and from the other four bearings. This entails the following operation sequence:—

(1) Finish grind front end bearing on a Landis machine.
(2) Finish grind centre bearing on a Churchill "Hydrauto" machine.
(3) Finish grind flange and bearing on a Churchill machine.
(4) Finish grind four intermediate bearings on a Landis machine.
(5) Finish grind flange diameter and back on a Churchill machine.
(6) Finish grind timing wheel end diameter on a Landis machine.

For all these operations the crankshafts are mounted on fixed centres. Remarkably close dimensional tolerances and a very high degree of surface finish are obtained by conventional grinding methods. An important factor in maintaining quality is that all grinding wheels are mounted on Shardlow wheel-plates. These comprise a pair of interlocking discs, one of which is dished to hold balance weights that are adjustable for peripheral position. Every wheel is accurately balanced before use.

After these grinding operations, the crankshaft is transferred to an Archdale horizontal multi-spindle drilling machine, Fig. 18, on which eight holes in the flange are drilled and counterbored. A trunnion-mounted four-station work fixture is used. The first station is for loading and unloading; at the second, eight holes are drilled; at the third, six holes are counterbored; and at the fourth, two holes are counterbored. From the drilling machine the work is transferred to a four-spindle Archdale horizontal tapper, on which four flange holes are tapped simultaneously. Following this, the flange dowel holes are finished to a diametral tolerance of 0·0005 in.

Fig. 17. Tocco hardening plant with trolley track for work transfer from station to station

A Landis machine, see Fig. 19, is used for finish grinding the crank pins. The shaft is mounted in air-operated chucks on workheads at either end of the machine. Each chuck is bored off-centre by an amount equal to the throw of the shaft. Correct radial location of the pin to be ground is given by a hardened plunger in the left hand chuck which registers in the appropriate hole in the flange of the shaft. Longitudinal location is obtained through a master spacing bar on the table. It regulates the lateral position of the table by engaging with a hydraulically loaded plunger on the machine bed.

In operation, the wheel is advanced by quick hydraulic motion and is then automatically slowed to feed rate to grind the collar faces. The lubricant is switched on just before the wheel contacts the pin diameter. While this diameter is being ground, the operator is assisted by different lights denoting the various stages of the feed. At a pre-determined size, the sizing gauge is placed on the crankpin; this automatically brings the steady rest into position while the feed is again reduced for the final sizing. When the final size is reached, an electrical circuit coupled to the sizing device, causes the wheelhead to retract. A full width wheel is used. It is dressed frequently. All the actions of this machine are indicated on a visible colour panel.

But little further machining is necessary, and before it is carried out the crankshaft is tested for cracks on Metropolitan Vickers magnetic equipment. Then a keyway is milled in the end of the shaft to take the timing wheel, the flange end is bored and recessed, and finally the ball race bore in the flange end is finished.

This last operation is interesting. It is carried out on a Reed Prentice lathe which has been specially adapted, see Fig. 20. A half-chuck with an air-operated extending centre is attached to the headstock and a large bracket fitted on the bedplate carries a roller race with an inserted ring that holds an adaptor to fit the flange of the crankshaft. Several interchangeable adaptors are available to suit different shafts.

The crankshaft is mounted with the timing wheel end in the chuck and the flange on six hardened steel pegs in the adaptor. An air-operated centre locks the flange in the adaptor. Work rotation is provided by a driving dog on the chuck. Two-directional clock gauge control of the movement of the Wimet tool is provided. The tool is hand traversed. Its transverse position is indicated to 0·0001 in on a dial on the saddle, which has its plunger actuated by a gauge block on the cross slide. Longitudinal traverse of the tool is recorded to 0·001 in on a dial attached to the machine bed.

At this stage, the shaft is again tested for "true," and if necessary, correction is made on a Mills press. Balancing is then carried out on one of the several types of balancing machines that are installed. The balancing operation calls for no comment other than that the utmost care is taken to ensure that the crankshaft is statically and dynamically in balance within very fine limits.

The final operation is to polish the bearings and pins on a special machine that has been developed by the Company. This machine is shown in Fig. 21. The crankshaft is loaded between a driven faceplate, mounted on the headstock and adapted to take the flange, and an adaptor that fits over both the timing wheel end of the shaft

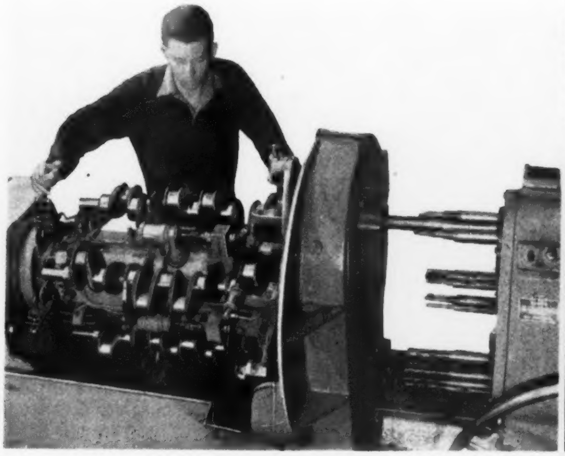

Fig. 18. Archdale multi-spindle drilling machine

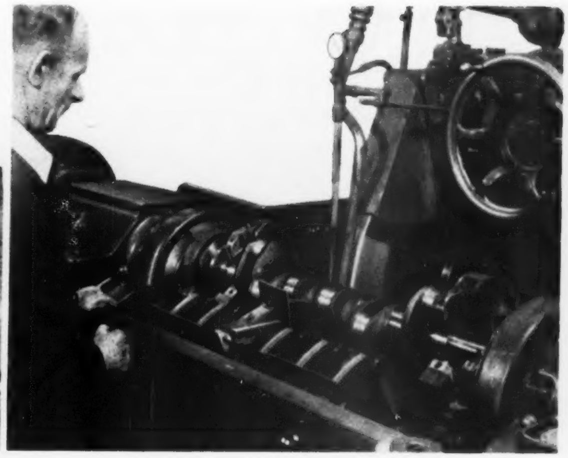

Fig. 19. Finish grinding crankpins on a Landis machine

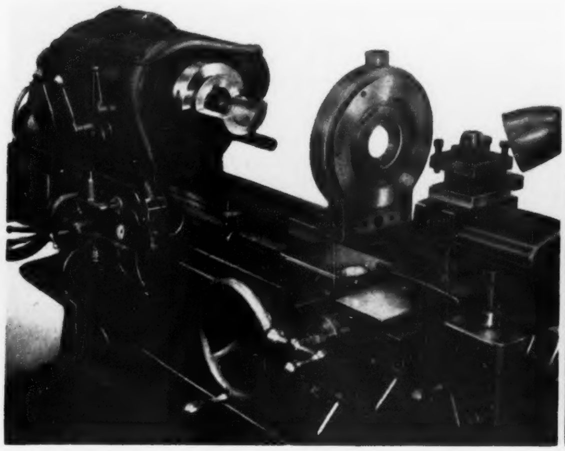

Fig. 20. Reed-Prentice lathe for finishing the ball race bore in the flange end of the crankshaft

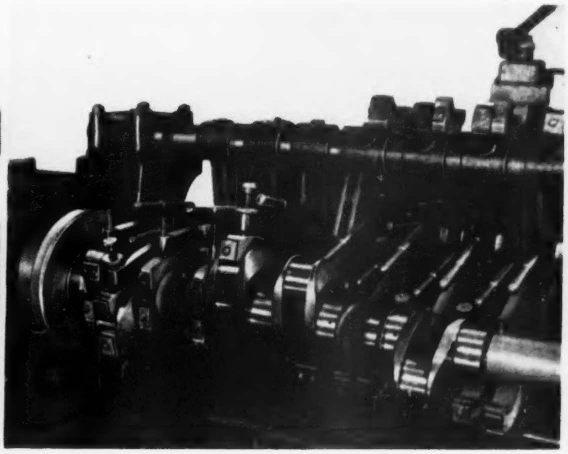

Fig. 21. A special Shardlow machine for polishing all the bearings and crankpins

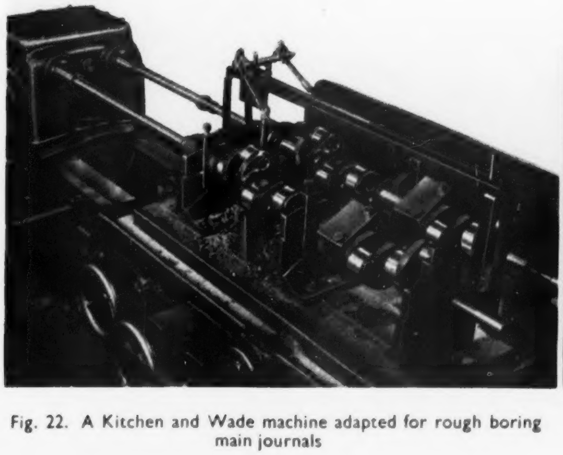

Fig. 22. A Kitchen and Wade machine adapted for rough boring main journals

Fig. 23. Contour grinding web contours on a Churchill machine with twin wheels

Fig. 24. Four-cabinet Tocco plant for hardening pins and bearings

and the spring-loaded centre in the machine tailstock. The headstock has a positive longitudinal movement of ¼ in and the tailstock is spring-loaded to accommodate the same amount of movement, while the crankshaft end is held by the adaptor.

Two long spindles, parallel with the work, overhang the back of the machine and serve as carriers for aluminium blocks with inserted felt pads. One spindle carries six rockers fitted to the laps by hinged joints so that they can reciprocate with the rotation of the crankpins. The other spindle has seven arms connected direct to the laps mounted on the bearings. The laps, which are smaller in width than the journals, are clipped in position; the drive is then started and the crankshaft rotates and also oscillates longitudinally. This machine produces a very high degree of surface finish through the combination of rotary and reciprocating movements.

#### Operations on other types of crankshafts

It is not possible to discuss the many interesting operations that are carried out on other types of crankshafts, but brief reference may be made to some. Many crankshafts have bored main journals, and one interesting set-up for rough boring main journals is shown in Fig. 22. The machine is a Kitchen and Wade horizontal duplex borer. Two shafts are bored simultaneously. The shafts are mounted side by side on a table that is mechanically traversed to bring the work to the tools which are mounted on extension bars. Guide bushes mounted at the front of the table provide the initial support for the boring bars. Further support for each bar is provided by a detachable jig that locates a hardened drill bush over the web face of the next journal to be drilled.

Another interesting operation is that of grinding the web contours on the Churchill machine illustrated in Fig. 23. On this machine the carriage (including the headstock, work and tailstock) is pivoted at either end and can be tilted transversely relatively to the wheel. This tilting motion is obtained through a cam in the headstock. The cam bears on a stationary follower attached to the fixed machine bed, and as it rotates it oscillates the carriage in accordance with the lift on its profile. In front of the cam, and arranged to rotate with it, is a face-plate with three equi-spaced slots used for indexing. The crankshaft is mounted between centres and is driven from the face-plate. Twin wheels allow two webs to be ground simultaneously.

Attention must also be drawn to a new four-cabinet Tocco machine for hardening pins and bearings. In this plant the shafts are held vertically and a number of pins and bearings are hardened successively without the operator needing to handle the crankshaft as he must do in the other Tocco plant. The cabinets have glass doors and the whole layout is particularly neat and orderly. This plant is shown in Fig. 24.

#### The toolroom

In any organization that produces work to fine limits a toolroom equipped for high quality work is an essential; it is needless to say that this applies to the toolroom of Ambrose Shardlow and Co., Ltd. Where it differs from many other toolrooms is in the manner in which its functions have been organized. In this department approximately 1,500 tools are reground every 24 hours and about 1,000 new cutting tools are produced every month.

Standardization of tools has been carried to great lengths. For example, turning tools are generally made of high speed steel and are 4 in long and can be ground to 1 in. Four cutting angles, 22, 15, 12 and 10 deg are used with a standard side rake of 3 deg. The side rake is not ground but is obtained by so designing the tool holder that the tool is at an angle to the work. Standard hardened blocks are kept in stock so that in an emergency a tool can be produced quickly. Standard jigs are also used to facilitate tool grinding.

So far as possible standard tool boxes are employed and incorporate simple but accurate means for adjusting tool height. They are made from high carbon steel, machined and ground on all surfaces. Where, as for the Maximatic multi-tool lathes, several tools are mounted in one box, the mounting is carried out in the toolroom. For long run orders, spare toolboxes with the tools in position are held in the toolroom ready for immediate issue as soon as they are needed.

#### Conclusion

Consideration of the Ambrose Shardlow and Co. Ltd. production organization as a whole clearly shows that product quality is always the overriding consideration. At the same time, the vital importance of production economy is never overlooked despite the difficulties occasioned by the widely varying demands for different types of crankshafts.

So far as possible the quality of the work is dependent upon the machine rather than on the operator, but the machine operators have always been encouraged to look upon themselves as more than mere machine minders. Every possible facility for the production of high quality work is provided for the operators, but they still retain responsibility for that quality. This is quite an important point in production economics since it allows the specified high standard to be maintained without the supervision of a large inspection staff.

## Helicopters

THROUGHOUT the world there is a growing interest in the helicopter. For a wide range of duties it can provide air transport under conditions quite impracticable for the normal fixed-wing aircraft which must take off and land at a relatively high speed. In the past decade production of helicopters has rapidly increased. To-day complete squadrons of helicopters are being operated under active service conditions with Allied military and naval forces.

Convinced of the importance of the helicopter in both military and civil spheres, our associate journal *Flight* is publishing on 23rd January a special issue on the subject. In this will be surveyed the whole field of rotating-wing aircraft and their power units, and special articles on design, application and operation will be included.

Copies can be obtained from all newsagents, or from *Flight*, Dorset House, Stamford St., London, S.E.1.

# WELDING RESEARCH

## *Some Methods and Equipment Employed at Abington by The British Welding Research Association*

THE headquarters and metallurgical laboratories of The British Welding Research Association are at Park Crescent, London, but its engineering research station is at Abington, near Cambridge. It is on the work carried out at the engineering laboratories that this article is based. Last June a new laboratory building was opened at Abington to house the equipment necessary for research into the fatigue properties of welded joints and structures. In passing, it is of interest to note that this building is the first to be designed in accordance with Professor J. F. Baker's plastic method and that as a result of the application of this method a considerable saving in steel has been accomplished.

It is clear that fatigue loading constitutes the most severe service condition for a welded joint, particularly in view of the fact that in most cases it is economically impracticable to completely eliminate the stress raisers that are almost inevitably introduced at such joints. Earlier investigations on this subject by the Association had to be carried out by the most economical means available. Therefore the resonance method had to be employed, but the limitations of this method became increasingly apparent as work proceeded. For instance, only equal tension and compression stresses can be applied. Moreover, the stress produced is a function of the design of the part under test and it is difficult to compare different structures such as, for example, riveted and welded units.

### Test machines

The limitations imposed by the resonance method were circumvented by the installation of a 100-ton Losenhausen fatigue testing machine, Fig. 1. This machine is designed to apply a pulsating load of up to 100 tons in either tension or compression. The alternating load range is 100 tons and may be applied anywhere within the limits of 100 tons compression and 100 tons tension. For example, the alternations mav be adjusted to give a range of 25 tons compression to 75 tons tension. For static tests, a load of up to 200 tons may be applied. The test specimen may be any length up to a maximum of 6 ft. Although there are other machines of this type in this country, none has a greater load capacity, and none is capable of taking so large a specimen.

The mounting of the Losenhausen machine is interesting. If it were solidly mounted in the normal manner in concrete, there would be a great danger that sooner or later foundations would crack, since the vibratory loads are so heavy and, unless unduly large foundations are employed, so concentrated. Accordingly, a specially designed flexible mounting is used to isolate the concrete base from the vibratory forces.

The machine is carried on four struts each of which is made up of six vertically positioned, bowed leaves not unlike those employed in a carriage spring, Fig. 2. Grooved blocks on the floor of the foundation pit and underneath the machine, locate the ends of the leaves. For levelling purposes, extra leaves of smaller cross sectional dimensions may be fitted between any of the four pairs of blocks. Further levelling adjustment may be effected by placing weights in channel-section trays welded on each side of the sprung mass.

Movement in a vertical direction is restricted by a square frame of channel section, horizontally mounted on four angle section legs welded to the base plate on the pit floor. Stop plates at each corner are bolted to the sprung part of the machine. They are equally disposed above and below the frame when the struts are in position and the installation level. Location against rotation about a vertical axis is provided by thin strips of high tensile steel placed horizontally, one along each side of the machine. Each is bolted at its centre to a block on the sprung mass, and at both ends to a block on the frame which is attached to the floor. These strips are very flexible, and so offer negligible resistance to vertical movement; but horizontal movement, and rotation about a vertical axis are prevented by tension between the centre and one end of the appropriate leaves. In the illustration showing these details, Fig. 3, the electric motor high up on the right is for raising and lowering the work table.

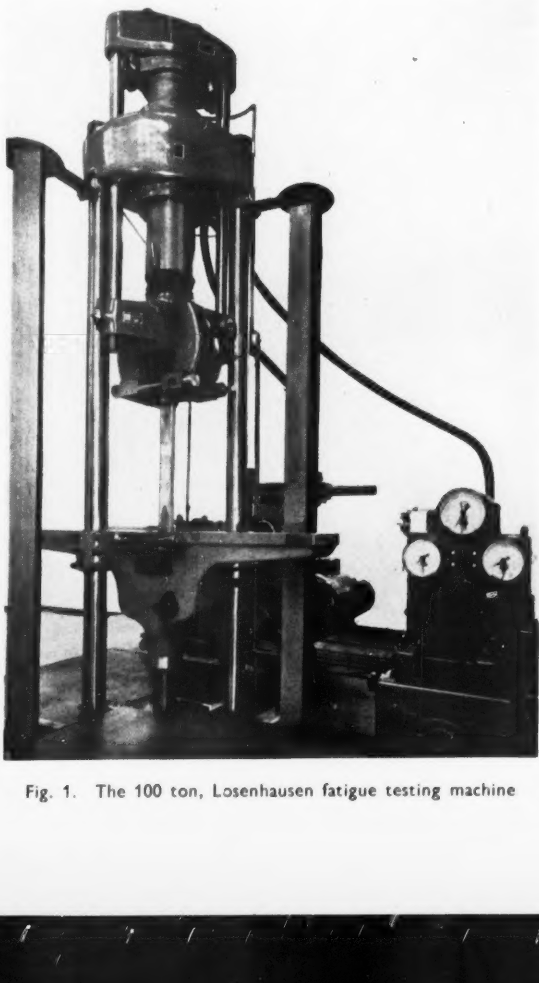

Fig. 1. The 100 ton, Losenhausen fatigue testing machine

Two more 100-ton testing machines of a different type have been installed for routine testing and to free the Losenhausen machine for work for which it is better suited. They are copies of a design by Professor W. M. Wilson of Illinois University, who supplied the information necessary for making the equipment. In the illustration, Fig. 4, they are shown in the course of erection. The 10 h.p. electric motor on the left serves the front machine, while that on the right serves the rear one.

The load is applied to the overhead beam by an eccentric which is adjustable to provide a range of amplitudes. The link between the eccentric and beam, not shown in the illustration, comprises an eye which fits on the eccentric and a screwed-in connecting rod capable of carrying tensile or compressive loads. This rod is mounted vertically with its upper end attached to a dynamometer. Another link connects the upper portion of the dynamometer to an extension plate bolted to the web of the overhead beam. The dynamometer is a solidly made, one-piece rectangular frame, suspended vertically

Fig. 2. Grooved blocks on the floor, and under the machine, locate the ends of the bowed struts

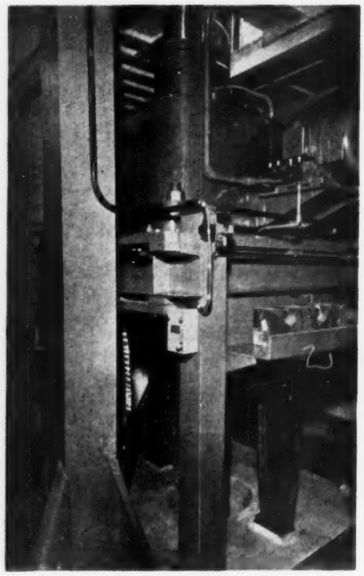

Fig. 3. The base of the Losenhausen machine is flexibly mounted on struts in a pit

and having an extensometer between its upper and lower members. Until the efficiency of the machine has been accurately determined, the strain must be measured by an extensometer on the test specimen. The other end of the overhead beam is overhung from a trunnion mounting, and carries two bearing heads, also trunnion mounted, one on each side. The specimen may be bolted between these two heads, and to two more heads on the lower beam.

A belt-driven, heavy flywheel is mounted on the end of the shaft that carries the eccentric. The function of the flywheel is to maintain a reasonably steady rate of rotation despite the varying load. The maximum speed is about 180 r.p.m., which is slow compared with the 600 c.p.m., of which the Losenhausen machine is capable. Another difference is that the Illinois machine runs at a constant strain cycle, whereas the Losenhausen machine gives a constant load cycle.

So far as the resonance method of fatigue testing is concerned, one of the most interesting features at the laboratory is the application of the method to longitudinal vibrations. This technique was developed by the staff, and useful results have been obtained. The larger machine, Fig. 5, gave a load of 30 tons in the test specimen, at a frequency of 4,000 c.p.m. The apparatus consisted of two large masses of 2½-3 tons clamped one at each end of the test specimen, and suspended by parallel links from overhead members. A Lanchester-type vibrator, mounted on one end and driven by an electric motor, supplied the excitation force, at a speed near the resonant frequency of longitudinal vibration of the system. Frequency control was maintained by electronic equipment fed with the voltage output from a strain gauge on the specimen.

A disadvantage of this application of the resonance method is that either a very long specimen, or very heavy masses must be employed to obtain frequencies and stresses of suitable values. However, before the larger machine was built, a smaller machine was made, to determine whether the method would be successful, and to find out what difficulties might be encountered, Fig. 6. In this case, a 6 ft. test piece was employed, and the masses were supported on top of vertical plates acting as parallel links.

At first sight it is surprising that the masses vibrate in anti-phase rather than in phase, but consideration of the dynamic principles of resonant vibration explains the phenomenon. It is apparent that unless the two masses are perfectly balanced relative to one another, a small in-phase longitudinal motion will take place.

**Light equipment**

Among the smaller pieces of apparatus employed at the laboratory are two interesting strain gauges. The first is the S.T.C.N. pneumatic extensometer, made by Solex (Gauges) Ltd., and the second the Johansson gauge. They both have advantages by comparison with the electrical strain gauges in common use. The S.T.C.N. apparatus, Fig. 7, which was developed by M. de Leiris of the French naval research laboratories, works on the well-known Solex air gauging principle. It has a gauge length of only 2 mm. This is a noteworthy feature, since the gauge may be used for measuring strains in relatively small localities where stress concentrations occur. Even the 2 mm. dimension is not small enough for all requirements, but it is much better than that of most mechanical or electrical instruments available. Although the gauge head is compact and fairly easily attached, particularly to ferrous test pieces, the water manometer with which it is used is bulky and not suitable for anything other than static tests in the laboratory. In the arrangement shown in the illustration, a magnet and bridge piece are employed to hold the head on a steel test piece.

The gauge head, Fig. 8, is of a simple design. Its principal components are two vertical legs pivoted together near their lower end by a very thin strip of metal. This type of pivot is used in preference to a pin because it is almost frictionless,

Fig. 4. The 100 ton Illinois, fatigue testing machine in the course of erection

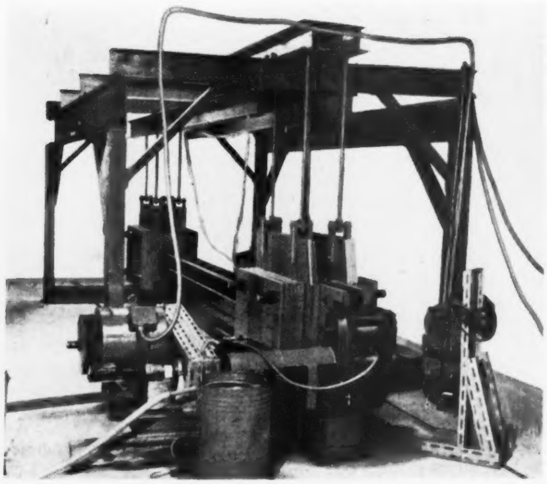

Fig. 5. A large, longitudinal stress, fatigue testing machine of the resonance type

The only resistance to the pivot action is the bending resistance of the strip, which is very small, and its internal frictional resistance which is even more minute. A short distance below the pivot are two gauge points, one on the end of each leg.

At the upper end of the head there is a jet, air from which impinges on a jet control plate. The jet is mounted on one leg and directed inwards towards the other. It is served by an adaptor to which is attached a short length of flexible rubber tube. At the other end of this tube there is a small air filter unit, which is connected to the water manometer by a length of heavier rubber tube. The jet control plate is circular, and is mounted on the inner end of an adjuster screw, with its axis in line with that of the jet carried in the other leg. A knurled head on the outer end of the screw, locked by a spring locking device, enables adjustment to be made by hand to the zero setting of the instrument. The function of the plate is to control the air flow from the jet. This flow will vary in proportion to the distance apart of the two legs, which is governed by variation of strain registered by the gauge points.

Fig. 7. The Solex strain measuring apparatus

A thumb screw is mounted in an adaptor immediately above the end of the leg carrying the jet. When tightened down, this screw bears on a conical cap carried on a thin strip of metal attached to the end of the other. leg. The cap then engages over a cone on the first leg, thereby setting the gauge length. This screw is turned back to its original position before the specimen is loaded for the test.

One of the principal difficulties associated with the use of this type of instrument in which the gauge points are so close together, is mounting the head satisfactorily. In this case, the head is held down by a pin passed transversely through one of the legs. It is positioned as low as possible below the level of the hinge point so as to minimize any overturning tendency. A forked leg at one end of a bridge-piece straddles the head and seats on this pin. At the other end of the bridge-piece, another forked leg rests on the test specimen. Both these legs are hinged so that they may be positioned at right angles to the surface above which they stand. In the illustration it will be seen that a rod, attached to a magnet which holds the apparatus down on the test piece, passes through the centre of the bridge piece which is held down by a compression spring between it and an adjuster nut on the end of the rod.

A constant level type water manometer is employed. It is interposed in the pipe line between the air supply and the gauge head. A fixed orifice is incorporated in this supply line, and a pipe to the constant level tank is connected at a point between the orifice and the air supply. This pipe is extended below the water in the tank and when its pressure head exceeds that between the lower end of the pipe and the water level, air from the supply is free to bubble out to the atmosphere. Provided that air is all the time bubbling out through the water, this device maintains a constant pressure in the supply line to the fixed orifice.

Another pipe connection is taken from the line between the orifice and the gauge head. This communicates with a simple U-tube, water manometer, which measures variation of pressure due to movement of the gauge points. The mechanical magnification between the points and the variable jet is 3:1, but the overall ratio at the manometer is 200,000:1.

The Johansson gauge, Fig. 9, is a very small dial instrument particularly suitable for making rapid surveys of stress in different parts of a structure. The gauge points are, as in the S.T.C.N. instrument, mounted on two legs. One of these points is easily removed and replaced by another to give a different gauge length, the alternatives being $\frac{1}{4}$ in or $\frac{7}{16}$ in. One of the legs forms the main pillar of the instrument, and carries the dial and the mechanism operating the pointer. The other leg is hinged to it near the dial, and has a zero adjusting screw adjacent to its lower end. This leg is part of a bell crank, but the hinge at the top is formed by two thin strips of metal, of such geometric arrangement as to cause the end of the upper lever of the bell crank to move over its short operating distance in a straight line instead of an arc.

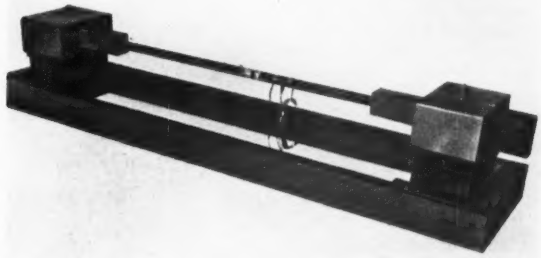

Fig. 6. A six foot test piece in a relatively small, longitudinal stress, fatigue testing machine

The upper lever is attached to the lower extremity of a twisted strip of phosphor bronze mounted vertically in, and attached at its upper end to, the main leg. The strip is twisted in opposite directions from each end; and at the centre a pointer is carried at right angles to it. Thus extension or relaxation caused by motion of the bell crank, is accompanied by a rotational movement of the pointer. An essential feature of the strip is that it is perforated along its centre. This greatly increases its efficiency, the ideal being to remove altogether the metal adjacent to its neutral axis. However, this ideal is unattainable, and the perforation method is a good compromise. One division on the scale is equivalent to 0·00001 in strain.

For mounting the gauge on the test

piece, a pin is passed through the main leg at a position below the dial. Over the ends of this pin is a small rubber cord, each end of which is attached to the arms of a stirrup held flat on the test piece. The other end of the stirrup is drilled and tapped to carry two screws that can be adjusted to form supports when the instrument is used on curved or other shaped surfaces. From the illustration it can be seen that one very convenient way of holding down the stirrup is to use a spring clip mounted on a magnet.

None of these instruments is suitable for use in the field. However, for this type of work the Research Association has a remarkably compact ten-channel strain gauge equipment, Fig. 10, made by Elliott Brothers (London) Ltd. Indications from all the strain gauge bridges may be projected simultaneously on the single cathode ray tube screen or, alternatively, they may be projected one at a time. A time base is available if required. For application in a vehicle on the road, the only equipment required in addition to that shown in

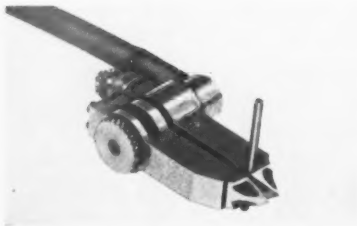

Fig. 8. The head of a Solex strain gauge

is clamped between knife edges at each end. A Lanchester-type vibrator is mounted outboard of the clamp at one end, and at the other a balance weight is secured to make the vibrating system symmetrical. The motor driving the vibrator is, of course, mounted separately on the floor.

In these tests the frequency is approximately 4,500 c.p.m., and the mode of vibration is such that nodes occur at the two supports. The machine is run at a speed slightly off resonance so as to avoid the peak of the amplitude response curve where

of diagonal lines are also drawn on the card. At the extremes of the up and down motion of the test piece, the card is stationary for an instant, so that the diagonal lines are visible. However, at all other times it is rapidly moving and they are more or less invisible. Thus, when the test piece is vibrating, a diamond shaped image can be seen. The greater the amplitude of vibration, the further apart will be the corners at each side of the diamond. It is a relatively simple matter to calculate, in turn, the amplitudes of vibration represented when the corners coincide with each of the vertical marks on the scale. These marks are of course visible under all conditions of vertical vibration. In this case, one division on the scale represented 0·050 in deflection.

Fatigue tests have also been carried out on butt welded pipe lines. Two conditions of loading have been investigated. The first is alternating bending, and the second pulsating pressure. It is possible that work on alternating bending, might be of interest in connection with butt welded axle tubes.

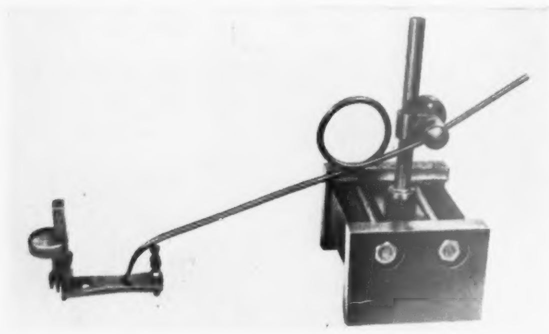

Fig. 9. The Johansson strain gauge is useful, in static tests, for rapid exploratory work

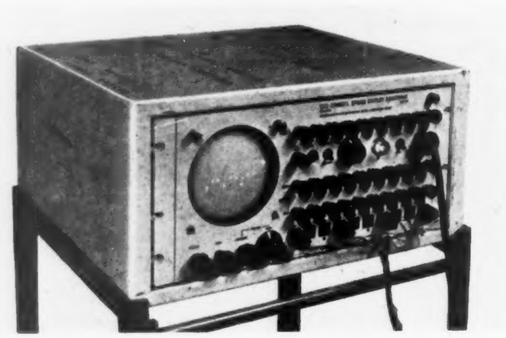

Fig. 10. Ten channel strain gauge equipment suitable for use in a vehicle on the road

the illustration, would be a battery and a generator. This extra equipment may be accommodated in the boot.

#### Work in progress

All the work in progress concerns welding, and is specialized. However, some is of direct concern to the motor industry. For instance, fatigue tests are being performed on cruciform bracing structures for chassis frames. This investigation is still in the early stages, and until it is completed the Association will not be in a position to issue any useful information on the subject.

Another section of the work at present being done concerns the fatigue strength of chassis side frames. Various sections have been prepared and welded by different methods, Fig. 11. Tests are now being carried out to determine their relative merits from the point of view of bending fatigue. No doubt the next step will be to perform torsion tests on similar sections.

A resonance method is employed for the bending tests, Fig. 12. The section

control would be difficult. A micrometer head contact arrangement midway between the ends of the test piece controls the frequency and cuts out the motor in the event of a large change of amplitude due to failure of the specimen.

An accurate check of the amplitude may be made visually by means of a simple device, which, in the illustration, can be seen attached at the centre of the test piece. It consists of a rectangular card with vertical lines marked on it to form a scale. A pair

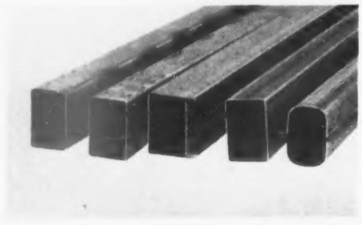

Fig. 11. A selection of chassis side frame welded sections prepared for fatigue tests

Other tests have been carried out on the fatigue strength of magnesium alloy welded structures, and on the influence of residual welding stresses on the fatigue strength of mild steel. With regard to the investigation of residual stresses, this has for a long time been recognized as an important problem which is difficult to investigate by experiment. The work is therefore a long term undertaking.

The residual welding stresses are simulated by a stress system produced by local induction heating of the surface of circular section bar specimens. A maximum surface temperature of 500 deg C is rapidly obtained. Since this heated zone is restrained against expansion, a small amount of plastic upset of the surface layer occurs. Then, on cooling, tensile stresses are induced. Specimens treated in this way are fatigue tested, and the results compared with those obtained from initially stress-free bars.

When high tensile steels are welded by the metal-arc process, hard zone cracking is liable to occur adjacent to the fused weld bead. This defect,

which is also known as cold underbead, base metal, or hydrogen cracking, does not occur in mild steel. The formation of these cracks is influenced by:—

(a) The composition of the steel.

(b) The type of electrode used.

(c) The rate of cooling in the heat affected zone.

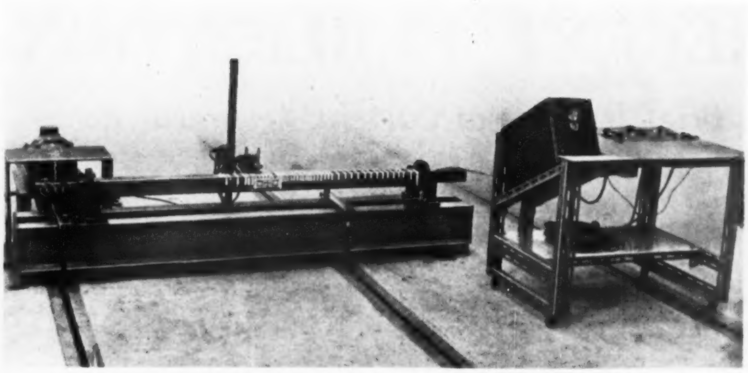

Fig. 12. The rig for bending fatigue tests on chassis side frames. The control table is on the right

In fact, for each combination of steel and electrode, there is a critical rate of cooling in the hard zone. Critical cooling rates have been determined for a number of different types of electrode used for welding manganese-molybdenum steel. These results indicate the marked superiority of the class 6, low-hydrogen electrode for welding high tensile steels. Further experiments have shown that under certain conditions, when steels are arc welded, a relationship exists between the incidence of hard zone cracking and the temperature for completion of austenite transformation during cooling. A test has been developed for assessing the weldability of high tensile steels without making a welding test.

Much of the work in the resistance welding section would be of interest to builders of light alloy coach bodies. For example, comparisons have been made between the strength of riveted aluminium alloy structures and spot welded ones. These tests have shown that in regard to both static and fatigue strength, the two forms of structure are substantially the same, provided the spot welding is properly executed with a condenser discharge type of welding machine.

The effect on the welds of a variety of pre-weld surface preparations has been determined. These preparations include simple degreasing, chemical cleaning and mechanical cleaning. For checking the surface resistance of light alloys before welding, the B.W.R.A., surface resistance meter has been developed. With this meter, a rapid and simple check may be made to ensure that the pre-welding cleaning operation has been carried out satisfactorily.

Another useful result obtained from experimental work has been the development of a slope control unit. In the past, it has been difficult to maintain with unfailing regularity the high currents necessary to give an effective spot welded joint. This is due to the fact that the area of contact between the pieces of metal to be joined is often very small owing to surface irregularity and imperfect cleaning. As a result, the points of contact have been grossly overheated at the instant the current was applied, and this has caused them to be blown away. The slope control unit was therefore developed to give a progressively increasing flow of current until the weld is completed. Such a unit has made easy, among other things, the hitherto difficult task of cross wire welding. It might be useful for obtaining consistency in the welding of chassisless structures.

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# INSTITUTION OF MECHANICAL ENGINEERS

## *Forthcoming Meetings of the Automobile Division*

The following meetings will be held during January:—

### NORTH-EASTERN CENTRE

*Wednesday, 21st January, 7.30 p.m. General Meeting in Chemistry Laboratory, The University, Leeds. Address by the Chairman of the Automobile Division, Mr. Maurice Platt, M.Eng., M.I.Mech.E. entitled "The Changing Practice of Automobile Engineering."*

### NORTH-WESTERN CENTRE

*Wednesday, 21st January, 7.15 p.m. General Meeting in the Walker Engineering Laboratories, The University, Liverpool. Paper: "Research and the Engineering Process, with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.), M.I.Mech.E. (Member of Council).*

### SCOTTISH CENTRE

*Monday, 19th January, 7.30 p.m. General Meeting in the Institute of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Some Problems Arising from the Wider Use of the Small Diesel Engine," by J. H. Pitchford, M.A., M.I.Mech.E.*

### WESTERN CENTRE

*Thursday, 29th January, 6.45 p.m. General Meeting at the Grand Hotel, Bristol. Paper: "Diesel Oil and Its Performance in the Engine," by Sydney Wightman, M.I.Mech.E.*

The following meetings will be held during February:—

### LONDON

*Tuesday, 10th February, 5.30 p.m. Joint Meeting with the Institution of Electrical Engineers. Paper: "Ignition Interference with Television Reception," by A. H. Ball, A.M.I.E.E., and W. Nethercot, M.A., B.Sc.*

### COVENTRY CENTRE

*Tuesday, 3rd February, 7.15 p.m. General Meeting in the Craven Arms Hotel, High Street. Paper: "Research and the Engineering Process, with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.), M.I.Mech.E. (Member of Council).*

### LUTON CENTRE

*Monday, 9th February, 7.30 p.m. General Meeting in the Town Hall, Assembly Rooms. Paper: "Research and the Engineering Process, with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.), M.I.Mech.E. (Member of Council).*

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## Aluminium Alloys

LAST year the Association of Light Alloy Refiners (ALAR) issued most useful data sheets detailing the revised numbers, related specifications and trade names of various British and American aluminium casting alloys. In order to make the information more readily accessible for rapid reference, a condensed version has been prepared and mounted on the two sides of a board for wall mounting.

In addition to the specification key it includes the chemical composition; physical, mechanical and general properties; and casting characteristics of the alloys. Copies are available to readers on application to ALAR Ltd., 3, Albemarle Street, London, W.1.

# RECENT PUBLICATIONS

## *Brief Reviews of Current Technical Books*

### Automotive Electrical Systems

*Edited by Irving Frazee and Earl L. Bedell.*

Chicago: AMERICAN TECHNICAL SOCIETY. Obtainable from The Technical Press Ltd., Gloucester Road, Kingston Hill, Surrey. 1952. $5\frac{1}{2} \times 8\frac{1}{4}$. 436 pp. Price 40s.

This is an American book and deals with American electrical systems, but the reviewer was left with the opinion that, in spite of the wealth of books dealing with this subject by British publishers and authors, there was still room for such a work as this. It is essentially a practical book for those working as electricians, or garage mechanics, and also for those who carry out their own repairs. Although most of it has been written by the editors, they have collaborated with many executives engaged in the practical, theoretical, and the teaching sides of the industry. No motor car repairer could fail to benefit from reading this book, and the differences in connection with American and British electrical equipment will not reduce the value of the information given by any great amount.

It tells how to diagnose the trouble in an organized manner, and how to correct it not merely by learning how to do it, but by learning why it should be done in that way. The student is encouraged to work from first principles. In each case full instructions, and the reasons why, are given. In fact, the method commonly adopted in motor car instruction books for trouble-shooting, under such headings as "Engine will not start," or "Engine starts but stops at once," and the subsequent procedure described for the benefit of the not too technical motorist, is somewhat similar to the methods adopted in some parts of this book.

The nine sections cover electricity and magnetism, storage batteries, generators, generator cut-outs and regulators, generating system trouble-shooting, starting systems, lighting systems, horns and sirens, instruments and gauges.

The book will be especially useful to those who are called upon at times to service American vehicles, for all the illustrations are of American components. Each chapter concludes with questions, together with page numbers on which answers may be found. A novel feature is the dust cover which unfolds to produce a starting system trouble-shooting chart.

### The Technique of Design

*By P. J. Wallace.*

London: SIR ISAAC PITMAN & SONS, LTD., Parker St., Kingsway, W.C.2. 1952. $4\frac{3}{4} \times 7\frac{1}{4}$. 103 pp. Price 12s. 0d.

Specialization from a comparatively early age is to-day the only hope for any student who aims to reach the top of any particular branch of a profession, and the amount of new knowledge that has to be added to that already known and which must be assimilated by any technical man is increasing rapidly every year. Specialization seems to be a necessary evil, for an evil it can be in certain cases. We do not know if the author of this book has considered the technique of design from this point of view, but it could be inferred from what he has to say. The author is very conscious of the fact, as indeed is industry generally, that students coming into industry from the universities, and even those who have sandwiched practical training into their studies, still have to acquire the technique of design. It is something that must be acquired, rather than learned, if they are ever successfully to make practical use of their academic training.

This is an unusual book for either the technical reader or the layman. Perhaps a more descriptive title for it would have been "The Mental Process of Achieving Successful Design Technique." The solution of any problem calls for mental treatment, but design problems in connection with new machines need organized mental procedures to facilitate solution. The theme of this book centres round the letters A.T.D.M., standing respectively for Analyse, Theorize, Delineate, and Modify. Before this, however, any designer should ask himself "Why do we want to do it anyway?" "What is it likely to cost?" and "What are the elements involved?" He is advised to bear in mind always the worst that might happen. As soon as possible he should put something down on paper to enable him to isolate the problems from the mass of detail.

The book abounds in sound practical advice and much of this is given simply and with some humour The book contains a deal of philosophical reasoning, and maybe a little sermonizing too, but it will be of interest to a wide variety of readers, especially if they are concerned with design.

To illustrate his method of going to work, the author has selected the design of a high-speed spinning rig, which develops from a first drawing (put something down on paper) to the final job in stages which show the mental technique as well as the mathematical calculations. These, obviously, must have been considerably more extensive than are shown, but can be found in any book on machine design. The measures advocated in this book are frequently disregarded or, at best, not fully appreciated.



### Materials Handling in Industry

London: BRITISH ELECTRICAL DEVELOPMENT ASSOCIATION, 2, Savoy Hill, W.C.2. 1952. $5\frac{1}{2} \times 8\frac{3}{4}$. 142 pp. Price 8s. 6d.

Short of a visit to the Mechanical Handling Exhibition and providing that the reader is only interested in electrically driven mechanical handling plant, there are few better ways of getting to know about the many types of equipment available, and their judicious selection, than by reading this book. This is the fourth in the Electricity and Productivity Series and its object is to show "how better materials handling can increase productivity and, at the same time, improve the workers' lot."

The subject is dealt with under five main sections: (1) Runways and Lifting Equipment, (2) Cranes, (3) Conveyors, (4) Floor Transport and Storage, (5) Miscellaneous Equipment. Each section is well illustrated with photographs and line drawings of the systems and equipment, with references in each case to the respective makers.

The information is in tabular form. Each sub-heading is numbered and is followed by two divisions of its text under respective titles "Use" and "General Details." For example, the Torsion Type Overhead Conveyor has for use "The handling of clothes on hangers in the garment industry, warehousing and packaging of light articles." The general details describe the illustration and state that the conveyor is very light, cheap and has certain advantages, which are given. Thus, in factual expression without padding, the book provides valuable information in small space. The 109 illustrations cover almost every known method of mechanical handling, and the products of about two dozen mechanical handling equipment manufacturers are included, most of these being photographs of the units at work in factories.

The book will certainly repay attention by any works executive whose products need to be handled during manufacture or must be passed from department to department.

# CURRENT PATENTS

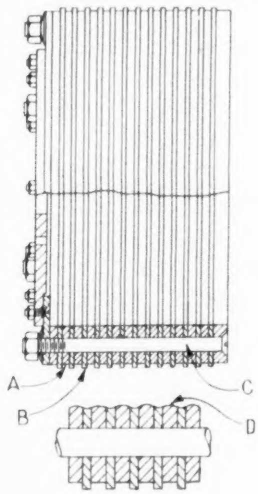
## *A Comprehensive Review of Recent Automobile Specifications*

### Laminated brake drum

TO reduce the size of a brake drum for a given braking effort and to obviate failure due to heat developed in service, a laminated brake drum having a plurality of metal rings, preferably of steel and copper alternately spaced, is suggested. The copper rings increase the heat conductivity of the drum and its ability to transfer heat generated at its braking surface to its outer periphery.

Steel rings A and copper rings B are assembled alternately and clamped by bolts C between annular end plates. In order to increase the dissipation of heat the copper rings have a greater radial depth than the steel rings, thus becoming in effect cooling fins.

As initially produced, the laminated braking surface will be cylindrical but the copper, being softer, will wear more quickly than the steel and the surface will rapidly become corrugated, as at D. This



No. 671299

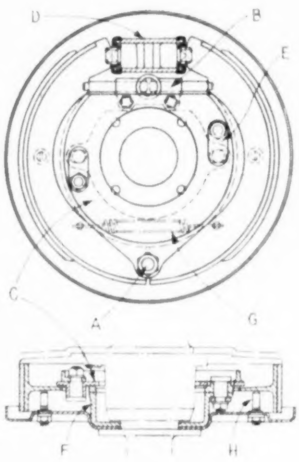
is claimed to be a desirable feature in that it increases the braking surface area. As a consequence the drum is expected to develop a greater braking effort for a given brake shoe pressure. *Patent No.* 671299. *Multi Ring Brake Drum Corporation (U.S.A.).*

### Equalizing Brake Shoe Pressure

TO avoid unequal wear on the linings of a self-energizing brake the shoes, with a common anchor pin A and an adjuster B, are mounted on a frame C which is permitted a limited floating movement substantially parallel to the line of action of the wheel cylinder D.

When the brake is applied the torque is taken by pairs of links E connecting the frame C with the back plate flange F. In operation, the leading shoe tends to rotate with the brake drum and causes a thrust to be transmitted to the end of the trailing shoe remote from the wheel cylinder, thus equalizing the work done by the two shoes.

Since its pistons can move un-

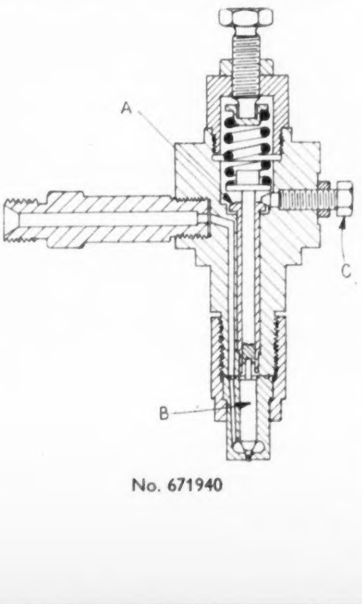


No. 671324

restrictedly, the wheel cylinder is rigidly attached to flange F. Reaction on the trailing shoe is taken by the appropriate tappet of adjuster B. The shoes are located by the constraint of spring G, which is offset from the plane of the webs and holds them in engagement with studs H mounted on the back plate. *Patent No.* 671324. *Girling Ltd.*

### Matching fuel injectors

HOWEVER accurately fuel injectors for compression-ignition engines are manufactured there are liable to be differences in the performance of individual components. Hitherto, it has been necessary, therefore, carefully to select a set of "reference" injectors for use on a fuel pump testing apparatus. The setting of multi-element pumps so that each element gives the same output would be facilitated if injectors could be accurately matched by means of simple adjustment. Furthermore, closely matched sets of injectors for particular engines can be



No. 671940

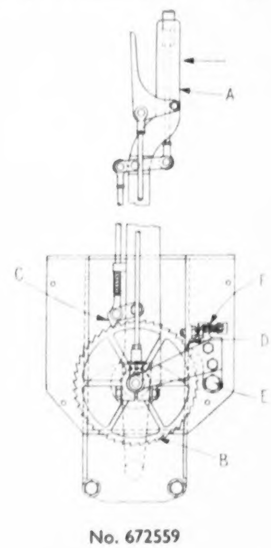
produced without a time-wasting process of selection.

In this injector the normal spring-loaded rod is surrounded by an axially adjustable sleeve A, the lower extremity of which forms an abutment limiting the lift of the nozzle pin B under fuel pressure. Screw C, having a conical end engaging the conical head of sleeve A, is adjusted to determine the axial location of the sleeve and consequently the maximum lift of the nozzle pin.

In an alternative construction the sleeve is threaded into the injector. The head of the sleeve is drilled with a number of radial holes which enable the position of the sleeve to be adjusted by a tommy-bar inserted through a slot milled in the body of the injector. *Patent No.* 671940. *L. Hartridge.*

### Multi-pull hand brake

EFFICIENT control of brake pressure during both application and release is possible, it is claimed, with this operating



No. 672559

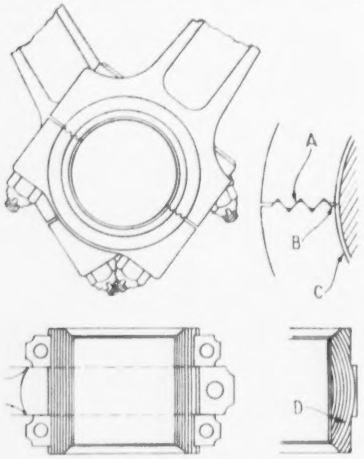
mechanism. Brake lever A is mounted coaxially with a ratchet wheel B which it operates by a spring-loaded driving pawl C releasable by a push-button. Check pawl D, pivoted at E and loaded by spring F, is controlled by a spoon handle mounted on lever A. An arm rigidly attached to pawl D is linked to the spoon at a point coincident with the ratchet wheel axis.

The brake is applied by pulling the lever several times with alternate return movements. When it is desired to release the brake the pawl C is lifted and the lever is brought to a convenient position and the pawl allowed to re-engage. After pressure has been applied to the lever the spoon is manipulated to release pawl D. The operator has then full control of the applied pressure through pawl C. If one forward movement of the lever is insufficient the operation is repeated. *Patent No.* 672559. *E.R.F. Ltd. and E. Sherratt.*

## Big-end bearings

IT is of utmost importance that a closed oil film under high hydraulic pressure be present between the crankpin and the running surface of a connecting rod big-end bearing. Diminution of pressure may occur should oil escape through the joints between the half-brasses and accordingly the joints are arranged remotely from the zone of high loading when the engine is running under maximum load conditions. When running at high speed under no load, however, major pressure may be exerted at different points and also at unfavourable angles detrimentally affecting the rigidity of the bearing. In the case of a V-type engine in which a rod is also mounted on the outer diameter of the brasses, these conditions are accentuated.

To obtain a relatively rigid bearing not susceptible to deformation, the brasses at their joint faces are furnished with axially aligned, interfitting serrations A. These have a triangular profile with an apex angle of approximately 90 deg. Adjacent the bearing surface are radially




No. 671197

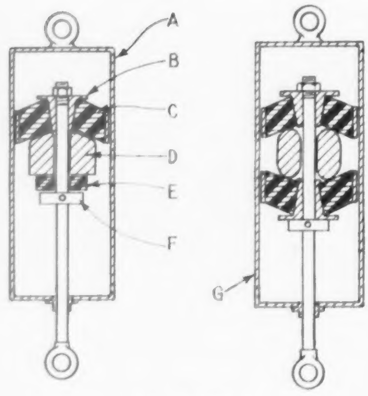
extending faces B and the lining C is suitably dimensioned so that on assembly the softer material is compressed to produce a running surface that, in effect, is jointless.

As an alternative, the serrations D may be arcuate to provide a still firmer connection between the brasses. The arcs are struck from a centre positioned far outside the bearing axis and so axial as well as radial displacement is prevented. *Patent No.* 671197. *K. Maybach (Germany).*

## Frictional shock absorber

AT any position within the working range the friction surfaces are devoid of effective engagement when at rest but, by means of a deformable member, are brought into engagement by relative movement. Springs or dash-pot devices are used only to supplement the frictional action.

In its simplest single-acting form the shock absorber comprises a closed cylinder A, through a gland in the end cover of which slides a piston rod carrying on a flanged sleeve B a frusto-conical, deformable rubber piston C. The periphery of the piston is sheathed with segments of friction material. Freely slidable on the rod below the piston is a metal block D, constituting an inertia




No. 672593

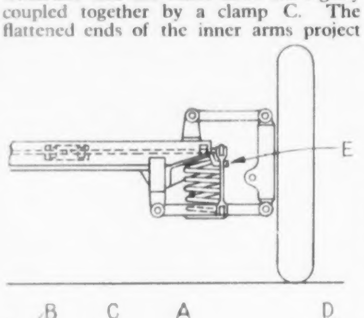
mass, located with an intervening rubber cushion E by collar F. Air in the cylinder is displaced past the piston and initial deformation of the piston is adjusted by means of the nut securing the sleeve B.

Modified constructions include the use of the cylinder space below the piston as a dash-pot, oil passing through an orifice in the end cover. Alternatively, both ends of the cylinder are open to atmosphere and a helical spring between the end cover and the piston supplements the effect of the inertia mass. At G is shown a double-acting component in which the inertia mass is positioned between opposed deformable pistons. *Patent No.* 672593. *Societa Applicazioni Gomma Anti-vibranti, S.A.G.A. (Italy).*

## Torsion rod stabilizer

THIS anti-roll bar is arranged within a hollow transverse member of a vehicle frame or an integral body structure, where it requires no additional space and is protected against damage. Further, it is utilized to augment the springing of the wheels and to check rebound.

In the example illustrated a dirigible wheel is mounted on a lower wishbone and an upper arm and sprung against the frame by a helical spring. The stabilizer comprises two torsion rods A, with ends bent to form lever arms, each mounted in two bearings in the transverse frame member B. Outer arms are linked to the wishbone and the inner arms are rigidly coupled together by a clamp C. The flattened ends of the inner arms project




No. 672402

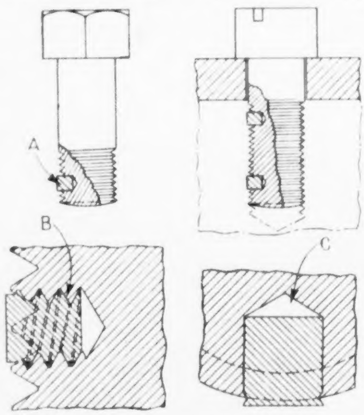
with clearance between upper and lower rubber cushions D.

In the event of equally large but oppositely directed movements of the road wheels, the rods A are torsionally stressed without the intervention of the intermediate abutments. When wheel movements are similarly directed there is no stressing of the rods as long as the inner arms move freely between the abutments. After engagement with one of the abutments, however, both rods are torsionally stressed and resist further wheel movement.

To check rebound movement, rubber cushions E mounted in brackets attached to the frame are engaged by the outer arms of the torsion rods. *Patent No.* 672402. *Daimler-Benz A.G. (Germany).*

## Locking device for screws

THIS idea exploits the fact that plastics of the polyamide condensation group, such as nylon, have the property of tending to return to their original shape after being deformed. The object of the invention is to furnish a means of locking




No. 671890

machine screws, bolts, studs, nuts, or the like. It is claimed that the nylon element retains its effectiveness after long-continued or repeated use.

As shown, a small blind hole is drilled in the threaded shank of the screw. The hole preferably terminates short of the axis of the shank so as to remove as small an amount of metal as possible. In the hole a plug A of Nylon 15 is secured. This plug has a smooth outer face which must lie above the thread root, and may project above the top of the thread.

Means must be provided for securing the plug in the hole and adhesives are regarded as unsatisfactory for the purpose. The property of nylon which is the basis of the invention is, therefore, again made use of. The nylon can be obtained in rod form of a size which will fit tightly, or may be slightly oversize and to facilitate insertion the rod should be extruded through a die or be rolled between pressure plates. Because nylon returns only gradually to its former shape, the plug can be inserted in the hole while still undersize and will recover to produce a tight fit. Alternatively, the hole may be tapped and the nylon plug screwed in, as at B.

It is recommended that a small air space C be left at the bottom of the blind hole so that the pressure of the entrapped air will assist the resilience of the nylon to exert a substantial pressure on the threads of the engaging screwed member. *Patent No.* 671890. *L. N. Brutus (U.S.A.).*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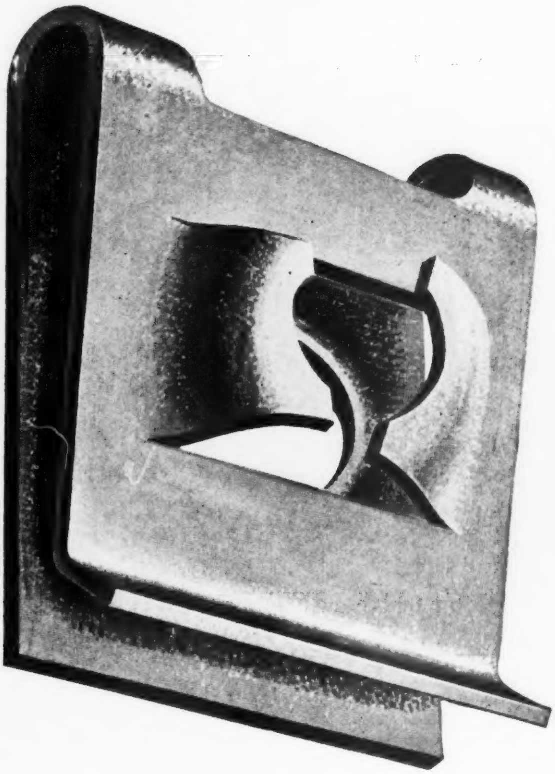



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**42 SNJ** Spire Speed Nuts are used to secure Radiator Shell; Steering Column Sealing Plate; Steering Column Blanking Rubber; Heater Blower Attachment Plate; Heater Blanking Piece; Bumper Valances; Heater Duct; Dash Cover Plate.

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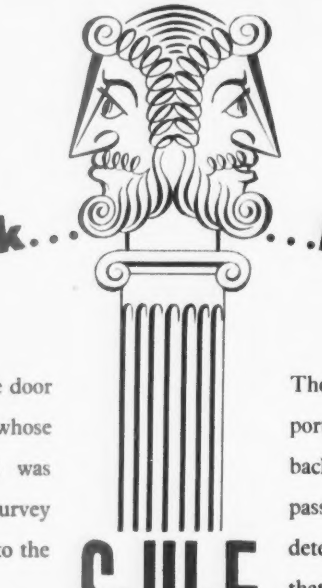
looking back... ...looking forward




The Roman God Janus (the keeper of the door between the old year and the new, from whose name the word January originated) was depicted as having two faces—one to survey the past and the other to look ahead into the future.

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* *This photograph was taken in the heat treatment shop of a leading manufacturer of bearings.*

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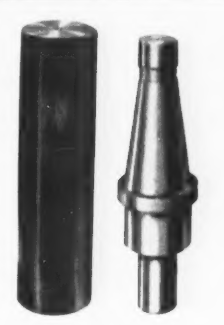

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*Depth of cut, 1/16 ins.*
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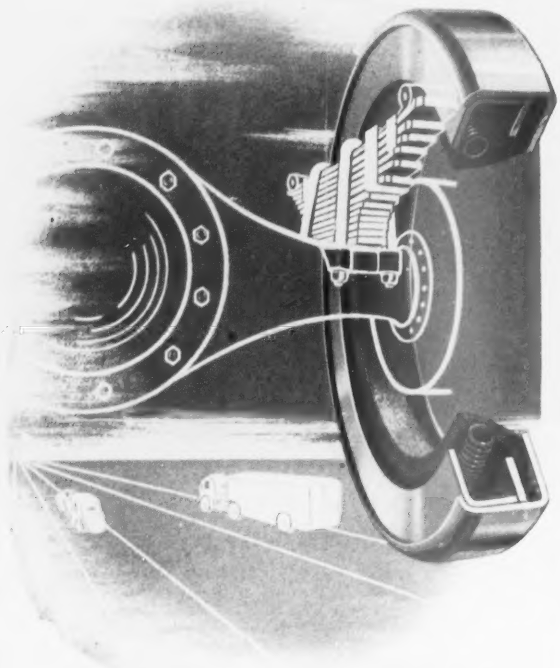

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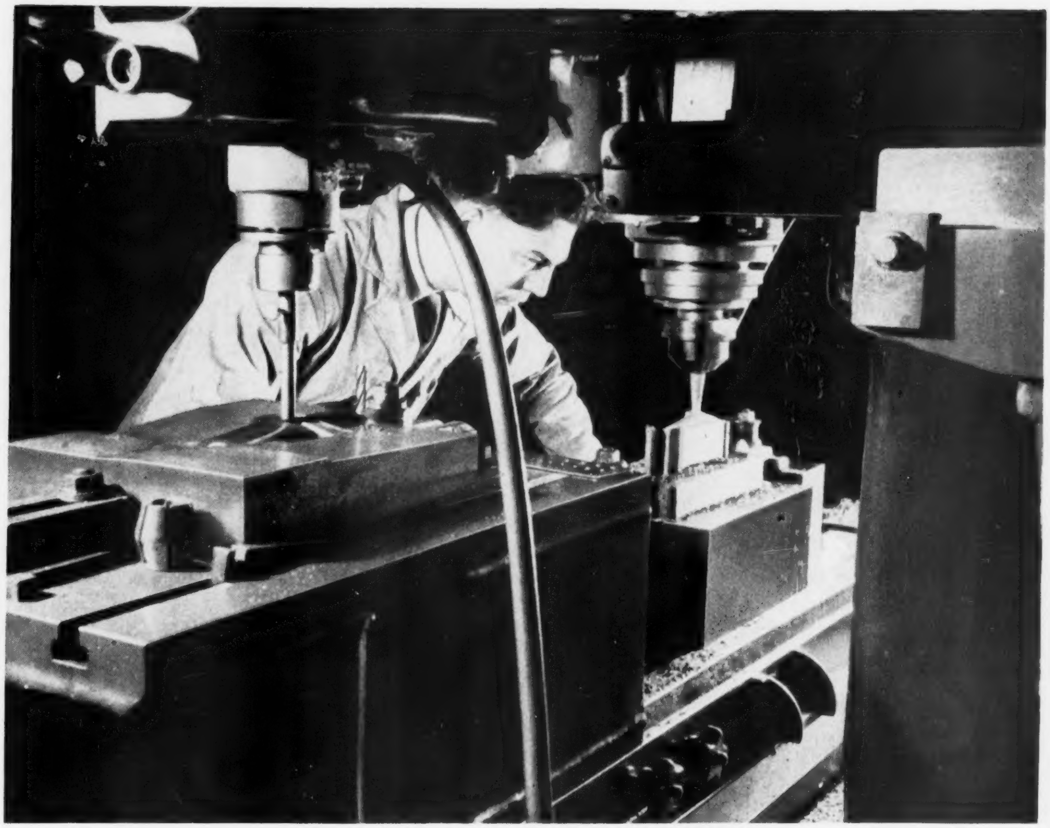

# Interesting
# . . . but tough

Heat-resisting steels are tough and not easily deformed even at high temperatures. Dies for drop-forging the elaborate shapes of gas turbine blades have a pretty hard job of work to do. Above are dies being machined at Doncasters on a profile copying machine. The lower illustration shows the beginning of marking-out a die block for the drop forging of connecting rods.
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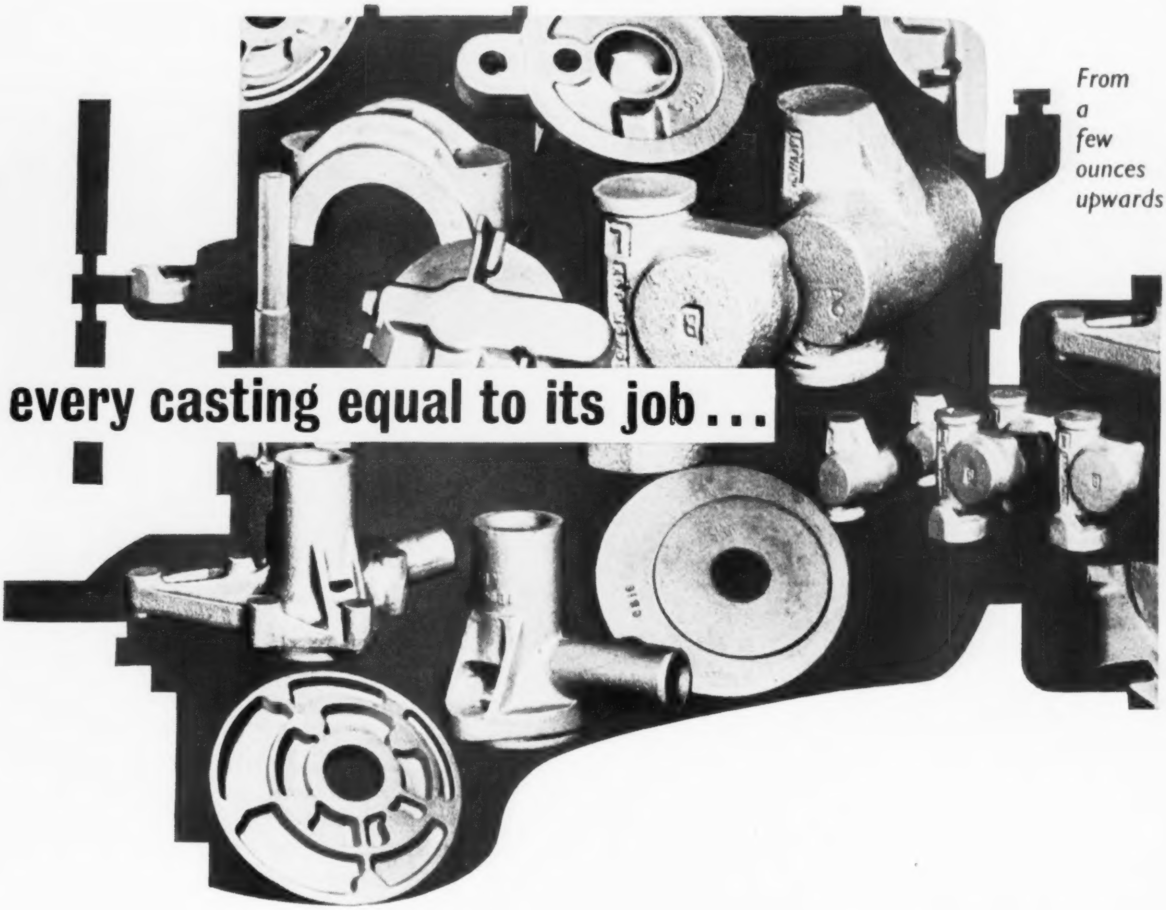


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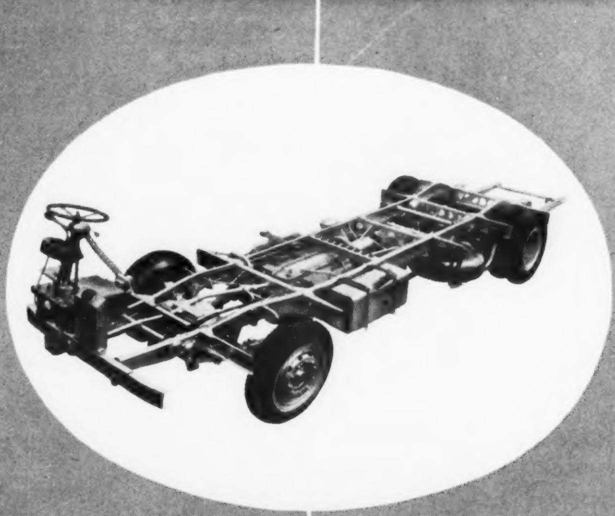



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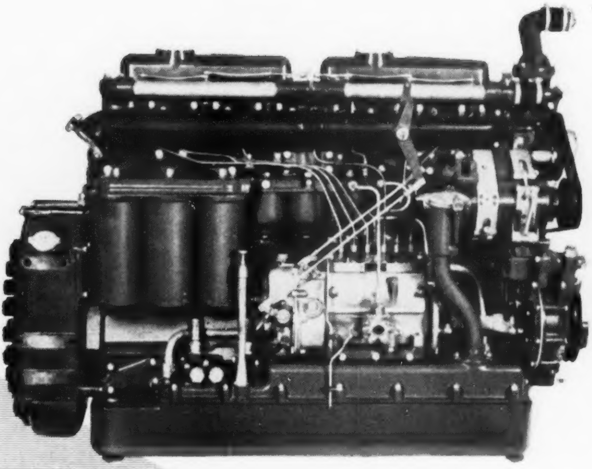



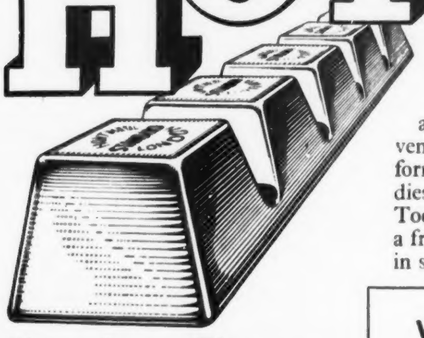



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